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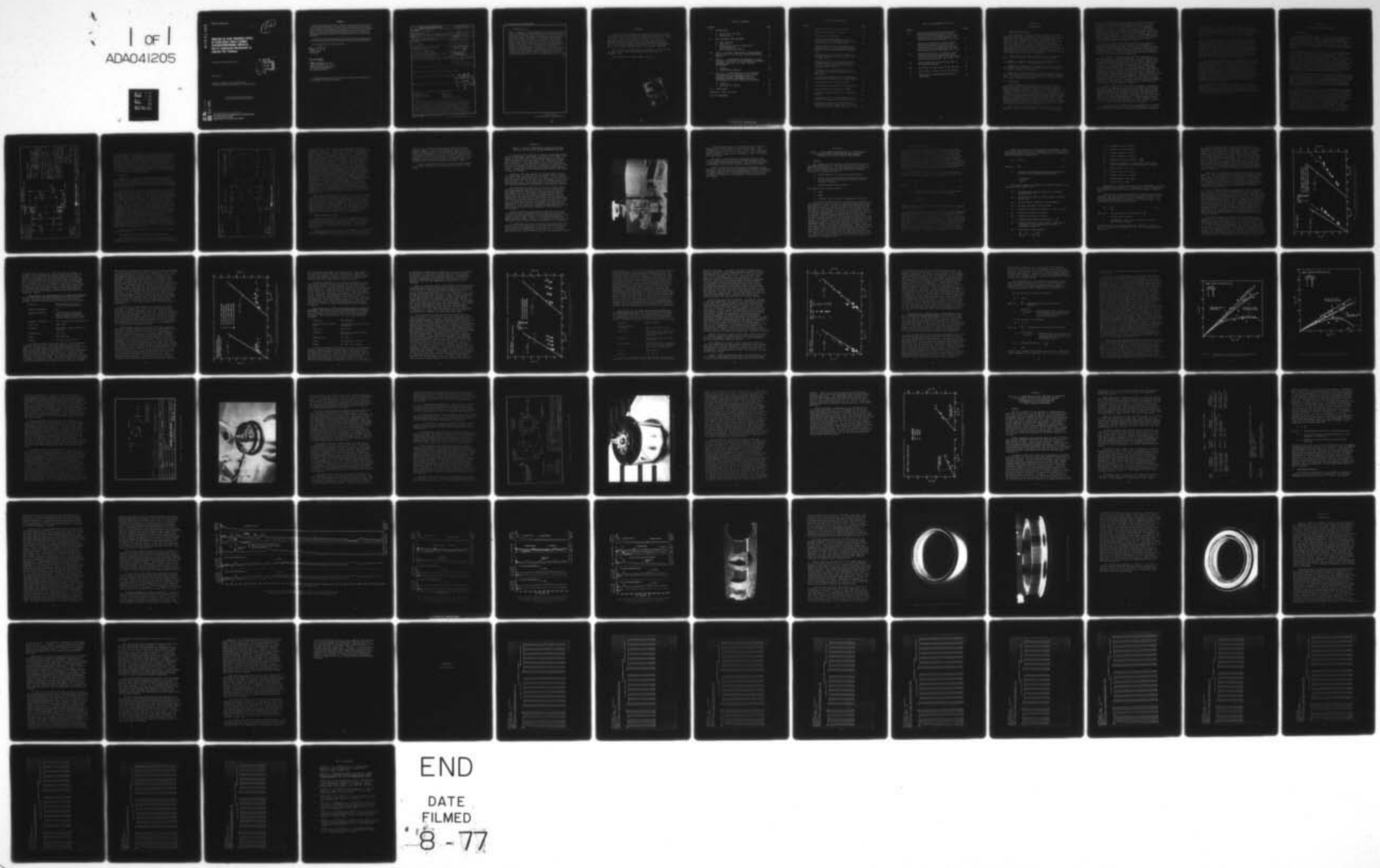

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AFML-TR-74-189, Part III

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ANALYSIS OF FILM THICKNESS EFFECT IN SLOW-SPEED LIGHTLY-LOADED ELASTOHYDRODYNAMIC CONTACTS
Part III. Experimental Measurement of Lubricant Film Thickness.

SOUTHWEST RESEARCH INSTITUTE

MARCH 1977

TECHNICAL REPORT AFML-TR-74-189, Part III
REPORT FOR PERIOD NOVEMBER 1975 – DECEMBER 1976

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AIR FORCE MATERIALS LABORATORY
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES
Air Force Systems Command
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This technical report has been reviewed and is approved.

Wayne E. Ward
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Project Engineer

FOR THE DIRECTOR

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SECURITY CLASSIFICATION OF THIS PAGE (When Data Entered)

REPORT DOCUMENTATION PAGE	READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER: AFML-TR-74-189, Part III — 2. GOVT ACCESSION NO.	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle): ANALYSIS OF FILM THICKNESS EFFECT IN SLOW-SPEED LIGHTLY-LOADED ELASTOHYDRODYNAMIC CONTACTS. Part III. Experimental Measurement of Lubricant Film Thickness.	5. TYPE OF REPORT & PERIOD COVERED: Final November 4, 1975 - December 4, 1976
	6. PERFORMING ORG. REPORT NUMBER: RS-651
7. AUTHOR(s): J. C. Tyler, H. J. Carper	8. CONTRACT OR GRANT NUMBER(s): F33615-76-C-5023
9. PERFORMING ORGANIZATION NAME AND ADDRESS: Southwest Research Institute, 8500 Culebra Road, San Antonio, Texas 78284	10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS: Project No. 7343, Task No. 734303
11. CONTROLLING OFFICE NAME AND ADDRESS: Air Force Materials Laboratory (MBT), Air Force Systems Command, Wright-Patterson Air Force Base, Ohio 45433	12. REPORT DATE: February 1977
	13. NUMBER OF PAGES: 78
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office): SwRI-RS-651	15. SECURITY CLASS. (of this report): UNCLASSIFIED
	15a. DECLASSIFICATION DOWNGRADING SCHEDULE

16. DISTRIBUTION STATEMENT (of this Report)

Approved for public release; distribution unlimited.

17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)

18. SUPPLEMENTARY NOTES

19. KEY WORDS (Continue on reverse side if necessary and identify by block number)

Bearings	Space
Accelerated Tests	Lubricants
Elastohydrodynamics	Lubrication
Vacuum	Ball-Piloted Retainer

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

This report presents a summary of a program conducted to study experimentally the factors contributing to the formation and maintenance of an adequate elastohydrodynamic oil film thickness in bearings operating in space hardware. Another objective of the study was to investigate the influence of the oil film thickness on bearing-lubricant life expectancy in typical despin mechanical assembly-type bearings operating in

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vacuum. Results of elastohydrodynamic film thickness measurements for DMA-type bearings operating in vacuum and having various ball-retainer materials as well as different retainer/bearing processing variables are presented and discussed. Also, the effects of frequency and extremes of temperature variation on these bearings are presented. Model studies using an EHD optical tester to perform fully flooded and starved lubrication experiments are also presented and discussed. The results from long-duration tests with DMA-type bearings having ball-piloted retainers and operating in vacuum are also presented. There were no long-term bearing failures and examination of the bearings after test termination reveals that substantially full EHD lubrication (not flooded, but separation of bearing surfaces) at the ball-race contacts apparently prevailed for the duration of the tests.

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FOREWORD

This report was prepared by Southwest Research Institute, 8500 Culebra Road, San Antonio, Texas, under Contract F33615-76-C-5023. The contract was initiated under Project No. 7343, "Aerospace Lubricants," Task No. 734303. The work was administered by the Lubricants and Tribology Branch, Nonmetallic Materials Division, Air Force Materials Laboratory, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio. The project engineer was Dr. W. E. Ward, AFML/MBT.

The report covers the period of November 4, 1975 through November 4, 1976, and was submitted by the authors in December 1976.

The contractor's report number is RS-651.

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SECTION I

INTRODUCTION

1. Objectives and Scope

The Statement of Work of Contract F33615-76-C-5023 specifies that the objectives of this program are: (a) to determine the factors contributing to and the effect of lubricant film thickness on long term performance in slow speed, lightly loaded rolling contact bearings (ball bearings) operating in a space (vacuum) environment, and (b) to develop relationships to explain lubricant film thickness and bearing performance in terms of the lubricant composition and systems parameters significant to long term performance in order to provide the foundation for the development of accelerated test methods.

Three major tasks were outlined in the Statement of Work. These tasks are identified herein as follows:

Task I — Design, fabrication, qualification, and delivery of an optical elastohydrodynamic tester.

Task II — Experimental measurements of film thickness in typical despin mechanical assembly bearings operating in a space (vacuum) environment.

Task III — Investigation of the influence of lubricant film thickness to ball-race composite surface roughness ratio on bearing-lubricant performance and life expectancy in a space (vacuum) environment.

2. Background

The successful long-term operation of any rotating mechanical equipment designed for use in space, such as despin mechanical assemblies (DMA's) and momentum wheels, is strongly dependent upon the performance of its bearing/lubricant system. Due to the high cost of launching a space vehicle, it is highly desirable that the bearings employed in stabilization components perform reliably without relubrication or maintenance for extended periods of 10-15 years. While successful continuous operation of more than 5 years has been achieved with some systems, many failures or abnormal performance attributed to the lubrication system have also occurred during this same time period.[1]

While there are many possible failure modes of space equipment that can be attributed to the bearing/lubricant system,[1] the study reported herein deals with the problem of the development and maintenance of adequate oil film thicknesses at the ball-race contacts of rolling-element bearings. In order to promote long life of a lubricated bearing assembly, it is important to maintain full elasto-hydrodynamic (EHD) lubrication conditions between the bearing components. Without full EHD oil films, surface contact and rubbing wear will occur. If operation in the boundary lubrication regime occurs for a significant portion of the mission, then excessive wear may occur, which will significantly increase the potential for bearing failure.

There are many factors which relate to the essential conditions necessary for the formation and maintenance of adequate EHD film thicknesses, in bearings employed in space hardware. These include lubricant properties, lubricant feed characteristics of retainer materials, lubricant processing of retainer/bearing, bearing design, bearing speed, bearing load, and bearing operating temperature.

As a means of estimating the EHD film thicknesses in bearings operating in space equipment, the classical theoretical film thickness equations[2,3] may be used. However, the theoretical equations apply strictly only under ideal conditions of perfectly smooth surfaces and fully flooded inlet conditions. Although bearing components are usually well finished, the surfaces have a finite roughness. Depending upon the ratio of the EHD film thickness to composite surface roughness, contact between asperities may occur, leading to rubbing wear. With respect to lubricant inlet conditions, there are several factors which contribute to lubricant starvation in bearings operating in space hardware. One factor contributing to possible starvation is the limited lubricant supply available to the bearings. Another factor promoting starvation is the continual wiping of the surfaces of the rolling elements by the rolling-element retainer.

Previous studies[4-6] have been conducted to determine the effect of lubricant starvation on the EHD film thickness. These experimental and analytical studies showed that as the supply of lubricant is reduced, the inlet lubricant boundary moves toward the Hertzian region, and the film thickness is reduced. As the inlet boundary approaches the Hertzian radius, the film thickness approaches zero. Although these studies succeeded in explaining the effect of inlet boundary on starvation, it is still not known how the inlet boundary position can be related to design and operating variables.

In summary, the present state of knowledge does not permit an accurate estimation of the oil film thickness in an operating bearing in terms of the design and operating variables known to influence the film thickness. Consequently, in order to evaluate whether or not an adequate oil film thickness prevails in the bearing so as to insure full EHD operation, experimental measurement of the film thickness in bearings operating in a simulated space environment is required.

In a previous study[7,8] sponsored by the Air Force Materials Laboratory (AFML), Southwest Research Institute (SwRI) developed a bearing race displacement measurement technique which provides a quantitative, accurate, and reproducible method of measuring the EHD film thicknesses formed in an operating bearing in vacuum. Experiments were performed on typical DMA bearings lubricated with various oils and operating in a space (vacuum) environment. The work reported herein is an extension of the previous work, and includes a determination of the effects of certain variables not evaluated in the previous study.

For the work reported herein, the experimental program involving measurement of the EHD film thicknesses in operating bearings was divided into two tasks. Task II involved short-term tests on typical DMA bearings operating in a vacuum to examine how EHD film thickness is affected by oil feed characteristics of different retainer materials, processing techniques used in applying lubricant to the retainer/bearing assembly, and frequency and extremes of environmental temperature variation. Complementing these studies with actual bearings, model tests were conducted in the SwRI optical EHD tester to study the factors which determine whether EHD contacts in a bearing operate in a flooded or starved condition.

In Task III, long-duration tests were conducted on bearings operating in vacuum. The objective of these tests was to generate experimental data relating the oil film thickness to bearing performance for the purpose of providing a realistic foundation for the development of accelerated tests for bearing life prediction.

SECTION II

TEST MATERIALS AND EQUIPMENT

1. Test Bearings

The test bearings selected for use in this program are of the design shown in Figure 1. Manufactured by the Marlin-Rockwell Company, these typical DMA bearings are ABEC-7 grade, angular-contact ball bearings with a counterbored inner race, 100 mm bore, and a contact angle of 26° ± 1°. The bearings have a ball complement of 19 balls, 15.9 mm (5/8 in.) in diameter. The rings and balls are made of 440C stainless steel.

The surface finish of the balls in all test bearings is approximately 0.025 μm (1 μin.), however, the surface finish of the races is not the same for all bearings. For the bearings used in the short-term tests of Task II, the surface finish of the races is the MRC "standard" finish of approximately 0.102 μm (4 μin.) transverse (across the grinding marks). All the bearings used in Task II had been manufactured previously for use in an earlier program.[8] However, only short-term tests had been conducted with these bearings during the earlier studies, so that they were suitable for reuse in this program. The bearings were thoroughly cleaned by BBRC before they were reprocessed with fresh lubricant.

For the long-term tests of Task III, the surface finish of the races is 0.403 μm (16 μin.) transverse. This relatively rough finish was used in order to generate a low ratio of the lubricant film thickness to ball-race composite surface roughness, for the purpose of determining the influence of this ratio on bearing-lubricant performance and life expectancy. The bearings for the Task III tests were new bearings fabricated by MRC for this program.

Three different materials and two different designs were employed for the test bearing retainers used in this program. The materials used for the retainers included porous nylon, porous polyimide, and phenolic, the latter being constructed of layers of cotton fabric impregnated with a thermosetting resin. All three materials are suitable for oil impregnation. The porous nylon material was manufactured by the Polymer Corporation and is identified as Nylasint 64HV-2. The porous polyimide material was manufactured by the Dixon Corporation and is identified as Meldin 9000. The phenolic material was manufactured by the Synthane-Taylor Corporation and is identified as Synthane Grade LBB. For the Task II tests, an outer-land-piloted retainer design was employed,

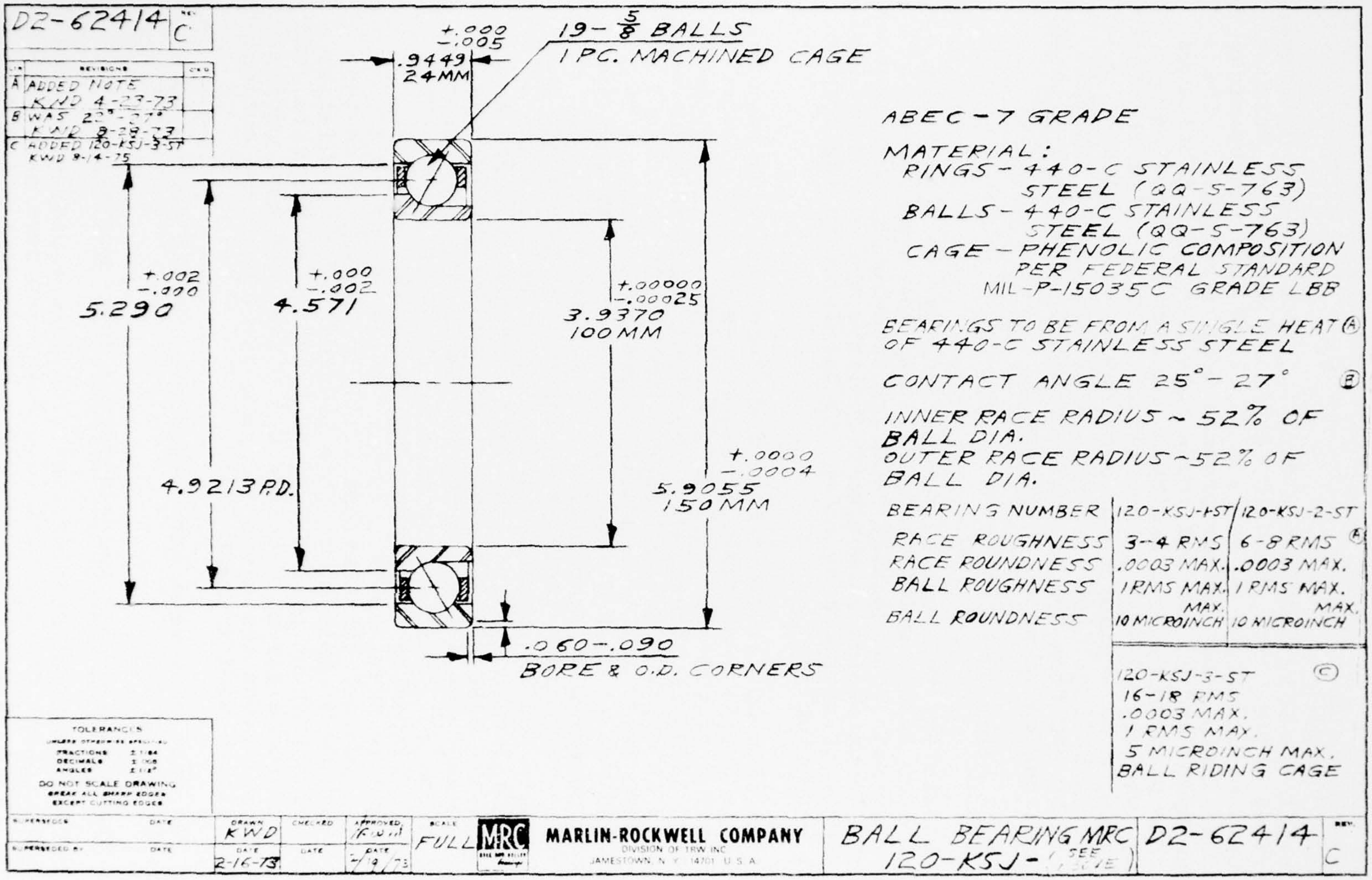


Figure 1. General Features of Test Bearings

and retainers were fabricated from all three of the materials described above. The phenolic retainers were those which had been manufactured by MRC and supplied with the bearings obtained during the earlier program mentioned above. The porous nylon and porous polyimide retainers were fabricated for this program by Ball Brothers Research Corporation (BBRC), who served as a subcontractor to SwRI. For the Task III tests, a ball-piloted retainer design was employed, and all of the ball-piloted retainers were fabricated for this program by MRC from the phenolic material. The ball-piloted retainer design is shown in Figure 2.

2. Test Oils

Basically, two test oils were included in this study. One is BBRC 36233, which is an oil that has been space-proven through considerable hardware experience; the other is Apiezon A containing an antiwear additive and an antioxidant. Two levels of antiwear additive concentration in the Apiezon A oil were studied.

The rheological properties for these oils have been presented in detail previously,[7] consequently only a brief description of the oils will be given here. The base stock for BBRC 36233 is Apiezon C, a low vapor pressure hydrocarbon oil used in vapor diffusion vacuum pumps. As determined at SwRI, BBRC 36233 has a kinematic viscosity of 105×10^{-6} m^2/s (105 cs) at 37.8 C (100 F). For an antiwear additive, the oil contains 5 percent of a concentrate of lead naphthenate. This concentrate is approximately 88 percent lead naphthenate and 12 percent straight-chain hydrocarbons with a lower boiling point than the lead naphthenate. For an antioxidant, BBRC 36233 contains 1.5 percent p,p'-dioctyldiphenylamine. The Apiezon A oil is also a low vapor pressure hydrocarbon oil, but is less viscous than Apiezon C. The kinematic viscosity of Apiezon A without additives, as determined at SwRI, is 28×10^{-6} m^2/s (28 cs) at 37.8 C (100 F). Two oil formulations utilizing Apiezon A as the base stock were employed in this program. One formulation contained 5 percent of the lead naphthenate concentrate, and the other formulation contained 0.5 percent of that additive. Both formulations contained 1.5 percent of the same antioxidant additive used in BBRC 36233.

The test oils were formulated and applied to the test bearings by BBRC.

3. Bearing Test Rigs and Associated Instrumentation

Four identical bearing test rigs were employed in this program. These bearing test rigs have also been described in detail previously,[7,8] thus only a brief description will be

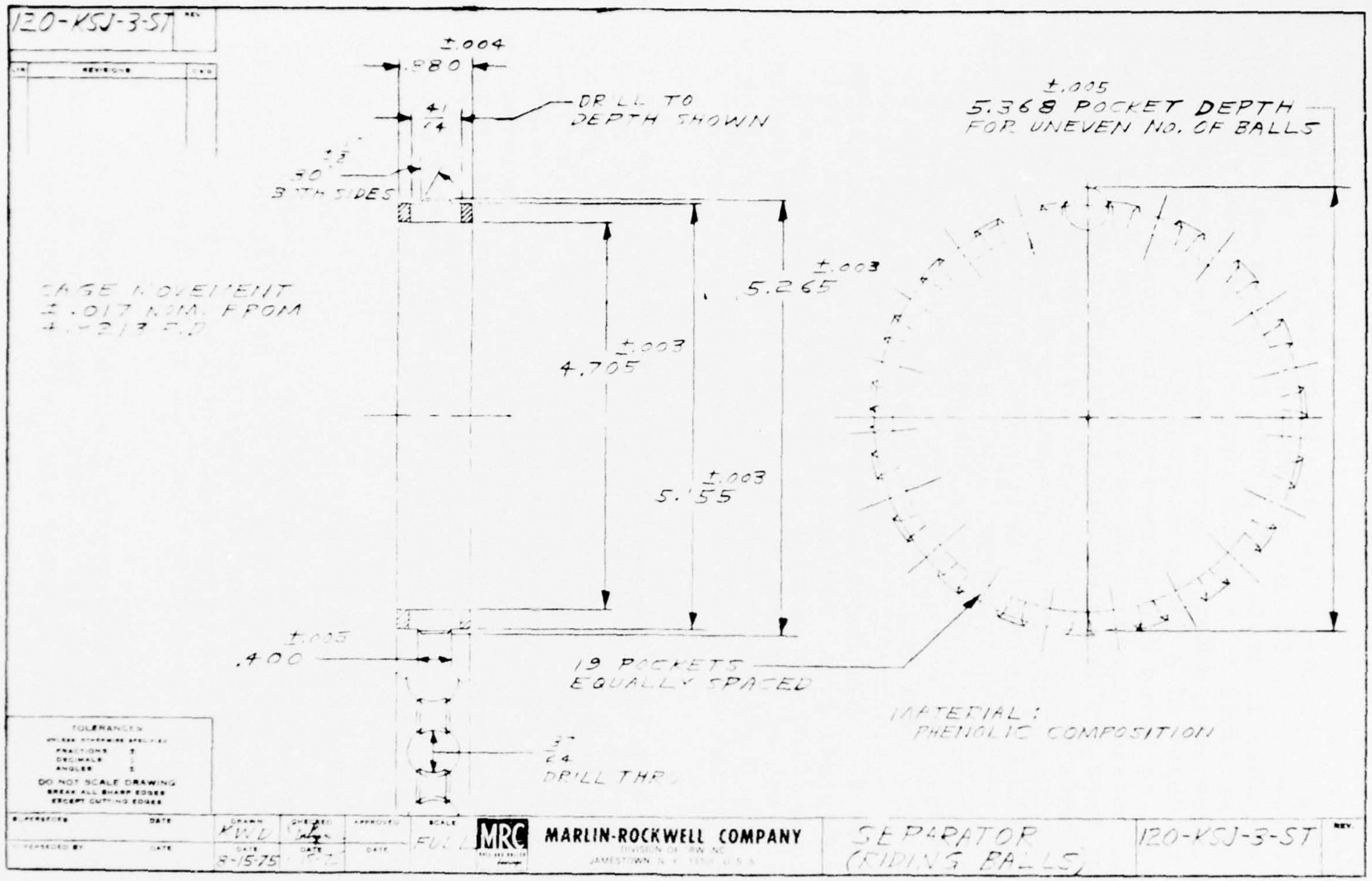


Figure 2. Ball-Piloted Retainer Design

presented here. Each bearing rig accommodates two test bearings, mounted on a single shaft and axially preloaded by means of a diaphragm. Preload can be set to any desired value between 222N (50 lb) and 890N (200 lb). The bearing-shaft assembly is contained within a cylindrical enclosure which is attached to a vacuum chamber to provide the simulated space environment. Rotary motion is imparted to the shaft by means of a magnetic coupling, the outer magnet of the coupling being driven by a variable speed DC motor mounted outside the bearing enclosure. The torque of each pair of bearings is determined by measuring the angular displacement between the driving and driven magnets, and relating this angular displacement to an angular displacement versus torque curve established by calibration. Motor speed is indicated by means of a magnetic pickup activated by a 60-tooth gear mounted on the motor shaft. Bearing temperatures are measured by means of thermocouples which contact the outer ring of each bearing. The forward bearing was held in the bearing cartridge shown in Figure 1 of Reference 8. This cartridge was attached to a diaphragm which provided axial preload to the test bearings. This forward bearing was closest to the vacuum facility and normally ran at a higher temperature than the aft bearing, especially during long-duration tests. The aft bearing was located closer to the drive motor as shown in Figure 1 of Reference 8.

The technique employed for measuring oil film thickness in these tests is described in detail in Reference 8. Basically, the technique involves measuring the axial displacement of the outer ring of the bearing which is elastically restrained by the loading diaphragm. This axial displacement is caused by the development of EHD films between the balls and races of the bearings. The measurement is made with a linear variable differential transformer (LVDT). Through the use of a computer program, the EHD film thicknesses are calculated from the axial displacement.

4. Bearing Retainer Models

To investigate the lubricant feed characteristics of various retainer materials, and also to study the effects of ball spacing on EHD film thickness behavior in a bearing, models of bearing retainers were employed. These models were tested in the SwRI optical EHD tester described in Reference 7.

The retainer models were designed by SwRI, and fabricated and impregnated with oil by BBRC. Two model designs were used. A single-ball design was used for the lubricant feed studies, and a multi-ball design was used for the ball spacing studies. Single ball models were made from the Nylasint 64HV-2 porous nylon, the Meldin 9000 porous polyimide, and the phenolic. Multi-ball models were made from the Nylasint 64HV-2 porous nylon, the Meldin 9000 porous polyimide, and from a porous nylon also manufactured by the Polymer Corporation and designated as Nylasint M-4. The Nylasint M-4 is less porous than the Nylasint 64HV-2.

A more detailed description of these retainer models is included in Section IV, along with the results of the tests.

SECTION III

TASK I — DESIGN, FABRICATION, QUALIFICATION, AND DELIVERY OF AN OPTICAL ELASTOHYDRODYNAMIC TESTER

As a part of this program, an optical elastohydrodynamic tester was designed, fabricated, and qualified at SwRI. The design of the AFML tester was patterned after an existing tester designed, built, and used in EHD lubrication experiments at SwRI.[9] The AFML tester is a versatile device, with which the oil film thickness profile in an EHD conjunction can be measured by optical interference techniques. The friction in the EHD conjunction can also be accurately measured. A photograph of the completed AFML optical EHD tester is shown in Figure 3.

Basically, the test section of the EHD tester consists of a steel ball in controlled rolling and/or sliding motion relative to two counterrotating flat disks. The two circular EHD contacts formed at the ball/disk interfaces are lubricated with the test oil. The 10.16-cm (4-in.) diameter disks are high-strength glass, the transparency feature being required for optical interference measurement of the EHD film profile.

The steel ball is driven by one of two variable speed DC motors supplied with the tester. With the two motors, a ball speed range of about 10 to 1,750 rpm can be achieved. The disks are driven by another variable speed DC motor. Through the use of a direction reversing gearbox, the two disks rotate at precisely the same speed, but in opposite directions. With the pulley and belting arrangement provided, the disk speed may be varied over a range of about 25 to 875 rpm. An electronic counter is provided with the tester for measurement of the ball and disk speeds. By changing the speed of the ball and disks, numerous combinations of rolling and sliding velocities may be obtained at the ball-disk conjunctions.

Load is applied to the ball-disk conjunctions through a rack and pinion arrangement which allows the upper disk to move with the application of calibrated dead weights. This loading system causes the ball to be squeezed between the upper and lower disks. Loads up to 68,950 N/cm^2 (100,000 psi), limited by local crazing of the glass disks, can be applied.

A gravity-feed type test oil system is supplied with the tester to provide lubrication to the ball-disk conjunctions. Conjunction inlet temperature of the test oil is measured with a miniature thermocouple that rides in the oil film on the rotating test ball slightly ahead of the conjunction inlet.

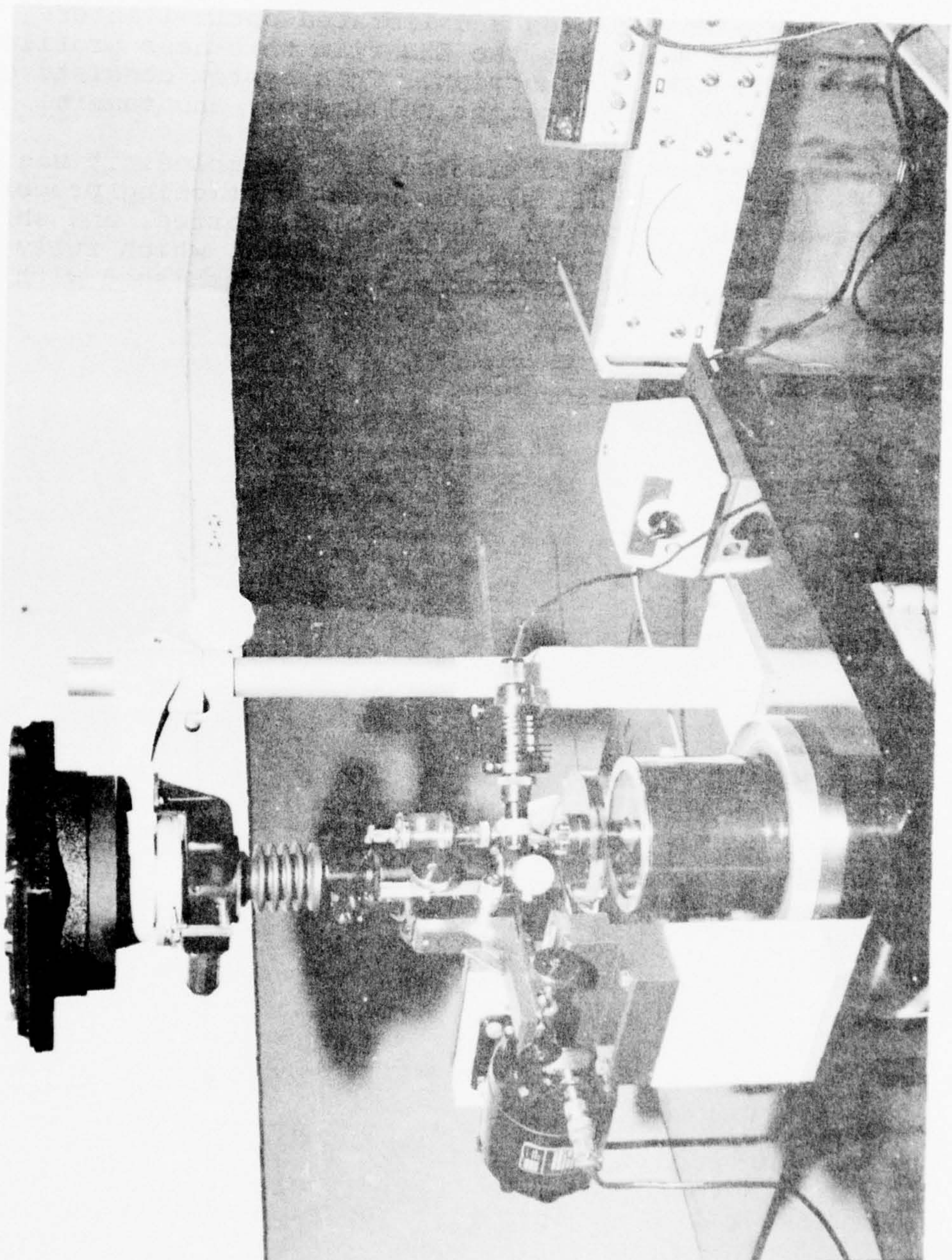


Figure 3. AFML Optical EHD Tester

A rotating torquemeter is provided with the tester to measure the friction torque in the contact zone. This torquemeter is installed in the ball drive shaft, and has a maximum capacity of 17.8 N-cm (25 oz-in.). The torquemeter was calibrated at SwRI, and a calibration curve is furnished with the tester.

The tester also features a calibrated optical interference system for measuring the EHD film thickness profile in the upper ball-disk conjunction. This system consists of a microscope, light source, light collimator, and camera.

After the tester was fabricated and assembled, it was operated to insure that all systems were functioning properly. The tester was then partially disassembled, carted, and shipped to AFML in August 1976. An operator's manual, which fully describes the tester and its operation, was furnished with the tester.

SECTION IV

TASK II — EXPERIMENTAL MEASUREMENTS OF FILM THICKNESS IN TYPICAL DESPIN MECHANICAL ASSEMBLY BEARINGS OPERATING IN A SPACE (VACUUM) ENVIRONMENT

1. General

The purpose of this task was to make quantitative film thickness measurements in the EHD conjunctions of typical DMA bearings operating in a simulated space (vacuum) environment. As previously outlined, the measurements include evaluation of parameters such as:

a. effectiveness of oil feed from the retainer

b. effect of retainer/bearing processing variations on the ability of the bearing to form a film of adequate thickness

c. flooded versus starved contact

and operational variables such as

d. load

e. speed

f. frequency and extremes of temperature variation

In the work conducted during previous years and reported in References 7 and 8, extensive film thickness measurements were made on various oils and oil-additive combinations using bearings operating in vacuum and also using the SwRI optical EHD tester. As reported in Reference 8, the effects of the variables of load, speed, and oil properties such as viscosity on EHD film thickness can be adequately correlated by the use of suitable dimensionless parameters which include these variables. Also as reported in Reference 8, no effects of the various additive packages on film thickness formation were observed. In addition, it is believed that the effects on film thickness of any batch-to-batch variation in base stock can be adequately handled by using the measured viscosity of the particular batch of oil under consideration. While there may very well be other effects of batch-to-batch variation in base stock and/or base stock-additive combinations in long-term performance, it is not believed that such effects can be observed in short-term tests such as those performed in Task II. Therefore, this variable was not included in these short-term tests, so that more emphasis could be placed on the variables in the above list.

2. Experimental Results

As mentioned previously in this report, a computer program was developed during an earlier study[8] for calculating the EHD film thicknesses in the various ball-race conjunctions in an angular contact bearing. The EHD film thicknesses are calculated from the bearing axial displacement, ΔL_y, measured by the LVDT system discussed earlier. This same computer program, which was described in detail in Reference 8, was also used to calculate the EHD film thickness for the DMA bearings tested in Task II. To correlate the experimentally-determined EHD film thickness data, dimensionless material-velocity-load parameters are used; these parameters are described below.

The best available expression for the central-region EHD film thickness in a flooded isothermal rectangular conjunction is due to Grubin[3] and is given by

$$H_c = 1.18\,\Sigma_G \tag{1}$$

where
$$H_c = \frac{h_c}{R} \tag{2}$$

Σ_G = Grubin's dimensionless material-velocity-load parameter for rectangular conjunctions

$$= \frac{G^{0.73}U_t^{0.73}}{W^{0.09}}$$

with the symbols defined after the minimum EHD film thickness equation given below.

While the central-region EHD film thickness is important to the present study, the minimum EHD film thickness is also extremely important. As is now well known, the oil film thickness profile in a rectangular conjunction is very nearly flat throughout, modified principally by a constriction in the exit region. This constriction, which is straight across the flow path for a rectangular conjunction, and almost straight for a high aspect ratio elliptic conjunction, results in a minimum oil film thickness within the conjunction, so that if surface-to-surface contact is to occur, it is apt to occur here first. Thus, the importance of predicting the minimum EHD film thickness in a bearing is evident.

Based upon theoretical analyses and considerable experimental data, Dowson[2] recently proposed the following equation for computing the minimum oil film thickness in a flooded isothermal rectangular conjunction:

$$H_m = 1.63\ \Sigma_D \tag{3}$$

where $$H_m = \frac{h_m}{R} \tag{4}$$

Σ_D = Dowson's dimensionless material-velocity-load parameter for rectangular conjunctions

$$= \frac{G^{0.54} U_t^{0.70}}{W^{0.13}}$$

The various symbols employed in the above equations, and throughout this report, are

H_c = dimensionless central-region film thickness, defined by Eq. (2)

H_m = dimensionless minimum film thickness, defined by Eq. (4)

h_c = central-region lubricant film thickness

h_m = minimum lubricant film thickness

R = equivalent radius of curvature = $(1/R_1 + 1/R_2)^{-1}$

R_1 = radius of curvature of body 1

R_2 = radius of curvature of body 2

G = dimensionless materials parameter = $\alpha_o \overset{*}{E}$

α_o = pressure-viscosity coefficient of lubricant at conjunction inlet temperature and near-atmospheric pressure

$\overset{*}{E}$ = equivalent elastic modulus

$$= 2\left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2}\right]^{-1}$$

ν_1 = Poisson's ratio of body 1

ν_2 = Poisson's ratio of body 2

E_1 = elastic modulus of body 1

E_2 = elastic modulus of body 2

U_t = dimensionless sum velocity = $\frac{\mu_o V_t}{\overset{*}{E}R}$

μ_o = absolute viscosity of lubricant at conjunction inlet temperature and near-atmospheric pressure

V_t = sum velocity = $V_1 + V_2$

V_1 = surface velocity of body 1

V_2 = surface velocity of body 2

W = dimensionless load = $\frac{w}{\overset{*}{E}R}$

w = load per unit width

Equation (3) is believed to be the best expression available for calculating the minimum film thickness for the rectangular or high aspect ratio elliptic conjunction with flooded, isothermal flow.

When the film thickness equations for rectangular conjunctions are used to calculate the film thickness in elliptic conjunctions, an equivalent load per unit width[9] is used. For a bearing with elliptic conjunctions, as in this study, the dimensionless load per ball is given by

$$W_e = \frac{w_e}{\overset{*}{E}R} \tag{5}$$

where w_e = equivalent unit load per ball = $\frac{3P}{4a}$

P = normal load per ball

a = semiwidth of major axis of contact ellipse at ball-raceway contact

and $\overset{*}{E}$ and R are as previously defined. Therefore, the W_e in Eq. (5) replaces W in both Eqs. (1) and (3) for angular contact bearings.

The above equations were presented and discussed in References 7 and 8 but were repeated here because they were employed in analyzing much of the Task II data. As will be seen, all of the Task II data presented in the following graphs with the exception of Test Series IV (model studies of EHD contacts under fully flooded and starved conditions) are compared with these equations. Also, as discussed on page 32, paragraph 3 of Reference 8, Eqs. (1) and (3) were used to calculate H_c and H_m at the base speed. These calculated values were then added to the film thicknesses determined from ΔLy to obtain H and H' plotted in the graphs. For all the film thickness measurements in vacuum presented in this report the base speed of the bearings was held constant at 25 rpm.

Since the four different conjunction films for each test condition were calculated from a single displacement measurement, ΔLy, they all followed similar patterns, although, the magnitudes of each would vary a few percent. The variance between inner and outer race film thickness are the result of a geometrical difference in the ball-race contacts,whereas the difference between aft- and forward-bearing contact film thickness values are a result of temperature differences between the two bearings.

Figure 4 shows typical film thickness values, calculated from the race displacement measurements, for the inner and outer contacts of the forward and aft bearings. The data were obtained prior to pumpdown for bearings treated with BBRC 36233. The dimensionless film thicknesses H and H' are plotted against the dimensionless parameters Σ_D and Σ_G respectively, which were defined and discussed earlier in this section of the report. As can be seen in the figure, the film thicknesses at all four locations display the same trends. Although there are slight load effects shown by the data at the larger dimensionless parameter values, a single straight line drawn through the experimental data for either H or H' would represent all the data very well. These data were presented previously in Reference 8 and are repeated here for demonstration and comparison purposes.

Since as indicated in Figure 4 the experimental film thickness data for the four different conjunction locations behave similarly, the data for all will not be presented. The aft bearing outer race contact data were arbitrarily selected for further analysis and comparison, and Task II, Test Series I, II, and III plotted data shown in the following graphs will be values for those contacts. For purposes of generality, all of the film thickness data are presented in dimensionless form. However, the dimensional value of film thickness may be obtained by multiplying the dimensionless film thickness H or H' by the equivalent radius at the ball-outer race contacts, R_o, which

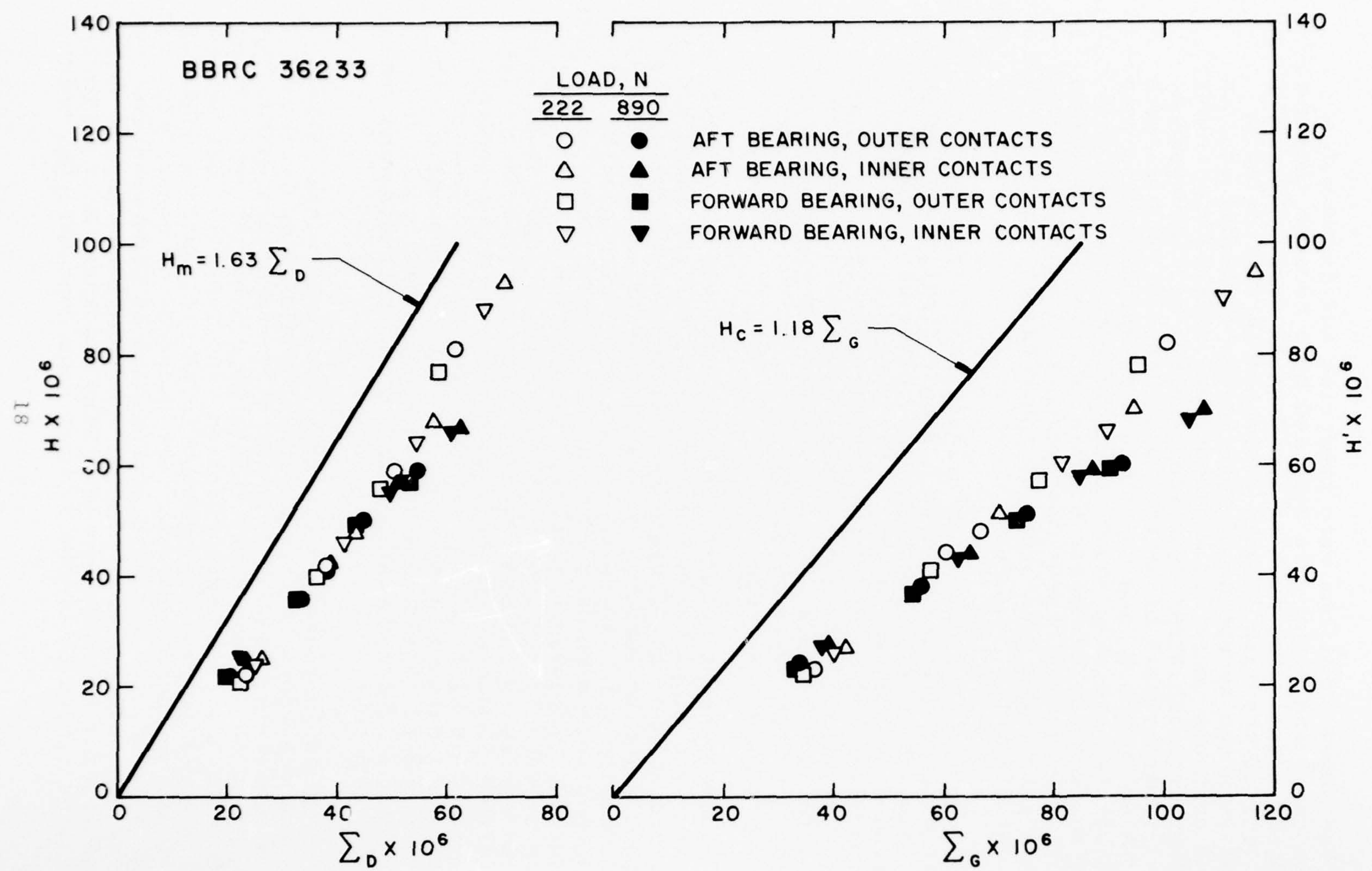


Figure 4. Dimensionless Prepumpdown Oil Film Thicknesses for Standard Bearings Having Thick Initial Film of BBRC 36233

is 8.84352 mm (0.34817 in.). The Test Series IV (model studies of EHD contacts under fully flooded and starved conditions) data were obtained using the SwRI optical tester and serve as support data for the other Task II test series. Since the data reduction and presentation were handled somewhat differently for Test Series IV, this will be discussed in more detail later in its respective section of the report. Similarly each test series of Task II along with plotted data representing that test series will be discussed separately in the following paragraphs. Highlights of these data will be pointed out and conclusions will be drawn later based on these test results.

Test Series I — Effectiveness of Oil Feed from Retainers. In this test series the effectiveness of oil feed from three retainer materials into EHD conjunctions of bearings was studied. The test conditions for the series were as follows:

Bearings:	MRC standard design with various retainer materials
Bearing treatment:	Dry
Retainer material:	Synthane-Taylor Grade LBB porous laminated phenolic, Meldin 9000 porous polyimide, Nylasint 64HV-2 porous nylon
Retainer treatment:	Fully impregnated with test oil
Test oil:	BBRC 36233
Pressure:	Equilibrium vapor pressure of test oil
Temperature:	25C (77F)
Load:	890N (200 lb)
Speed:	50, 100, 150, 200 rpm

For these tests the balls and races of the bearings were initially clean and dry. Each test employed a set of two bearings, and with each test a different retainer material was used. All of the bearing retainers were treated by Ball Brothers Research Corporation (BBRC) and were fully impregnated with the maximum amount of BBRC 36233 oil. BBRC also fabricated the porous polyimide and porous nylon retainers. The phenolic retainers were those originally supplied with the test bearings as received from MRC. The phenolic material represents a standard contemporary retainer material, while the polyimide and nylon retainers represent more advanced materials. BBRC data show that typically the phenolic held 7 percent by weight of the test oil, while the nylon and polyimide held 25 and 27 percent by weight of the test oil, respectively. Due to the short-duration nature of these tests, no reservoirs were installed in the bearing test chamber. Initially it was planned to include two load levels of 222 and 890N (50 and 200 lb) in these tests, but during the first test it was decided that using only one load was less likely to interfere with the oil transfer from the retainer to other bearing components. Also, less chance of contaminating the bearings was likely when using only one load, because the bearing test chamber had to be removed from the vacuum facility to change the load. It was thought that any bearing contamination or disturbing any oil transfer that might be occurring during these tests would be of more significance than in the other Task II tests where the bearing races and balls were coated with oil prior to testing. Since the bearings seemed to perform better at the higher load of 890N (200 lb), the remaining tests in this test series were limited to this one load level.

The reason for using initially dry bearings in these tests was to observe the effect of oil transfer from the retainers to the bearing balls and races. Since the balls and races initially contained no oil, any film that developed would be the result solely from feed of oil from the retainers to the bearing components. In each test, the bearings were first run slowly at approximately 35 rpm and the ΔLy measurement monitored to observe the development of any oil films as oil was transfered to balls and races. If an equilibrium film thickness condition was obtained, the measurements at the various desired speeds were made.

Unfortunately, neither the Grade LBB porous laminated phenolic or Meldin 9000 porous polyimide provided sufficient oil feed (transfer) even after 24 hours of operation to generate a measurable film. On the other hand, immediately after starting the test using the Nylasint 64HV-2 porous nylon retainers, a measurable ΔLy was obtained, but after several hours of operation a bearing "lockup" resulted while attempting to perform ΔLy measurements. The rig was disassembled, but no apparent cause of the "lockup" was found. Therefore, the rig was reassembled and the test again started and completed with no further problems. Comparisons of these data obtained for the bearings having Nylasint 64HV-2 porous nylon retainers with Dowson's Eq. (3) and Grubin's Eq. (1) are shown in Figure 5. It is apparent especially at higher values of Dowson's and Grubin's parameters, that there was not a sufficient amount of lubricant to provide either central-region or minimum film thicknesses near those predicted by the equations. On the other hand, even though flooded conjunctions apparently were not obtained, operation

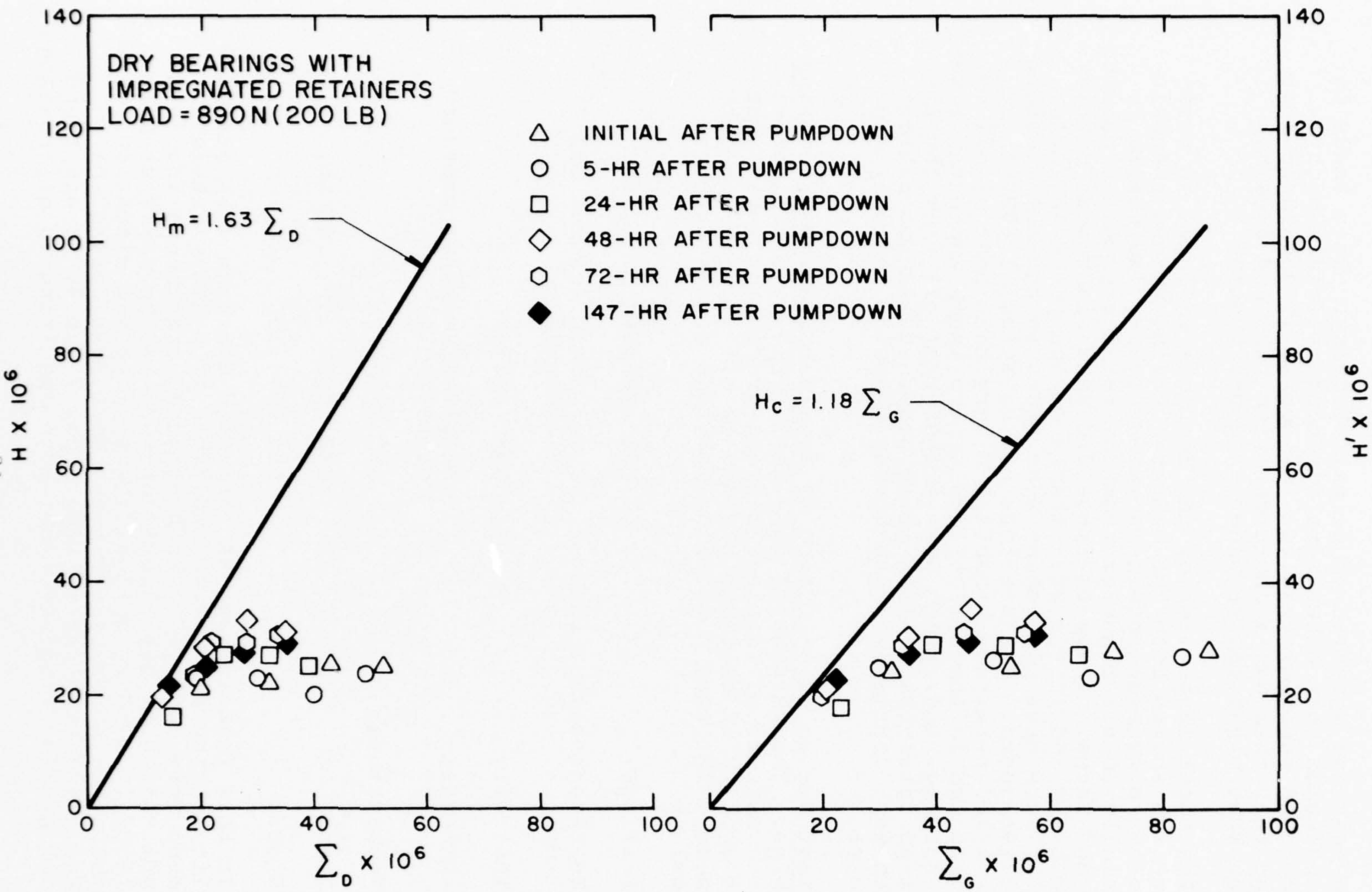


Figure 5. Dimensionless Oil Film Thickness for Standard Design Bearings Having Nylasint 64HV-2 Retainers Impregnated with BBRC 36233

of the bearings appeared to be satisfactory. This leads to the conclusion that other more advanced materials might provide even superior results. Of course it should be kept in mind that this was a short-duration test and does not predict what might happen under extended testing.

Post-test inspection of the bearings employed in this test series showed both the bearings having porous phenolic and porous polyimide retainers were very dry indicating essentially no oil transfer from the retainers to the bearing balls and races. These bearings did show black wear debris on the balls and in the wear tracks of the inner and outer rings. There were also shiny wear spots in the ball pockets of the retainers. The bearings having Nylasint 64HV-2 porous nylon retainers exhibited a very thin coat of oil on the balls and in the ball tracks of the inner and outer rings after testing, indicating some transfer of oil from the impregnated retainers. Also, these bearings did not exhibit as much wear debris as those having phenolic and polyimide retainers.

Test Series II — Effect of Retainer/Bearing Processing Variables. The purpose of this test series was to investigate the effects of retainer/bearing processing variations on the ability of bearings to form an oil film of adequate thickness. The test conditions for this series were as follows:

Bearings:	MRC standard
Bearing/retainer treatment:	2 very thin initial oil film thicknesses
Test oil:	BBRC 36233
Pressure:	Equilibrium vapor pressure of test oil
Temperature:	25C (77F)
Load:	222, 890N (50, 200 lb)
Speed:	50, 100, 150, 200 rpm

In this test series, MRC standard bearings with standard phenolic retainers and two conditions of bearing/retainer treatment were employed. Both of these treatments involved the application of less lubricant than is typically used in an attempt to produce a significantly starved situation. As has been discussed earlier in the work done during the previous years reported in References 7 and 8, it was found that for bearing/retainer treatments resulting in oil coatings on the bearing components of about 0.1 μm (4 μin.) or greater, no difference in the resulting EHD film thicknesses which developed in the bearings could be detected. Consequently, in order to provide an oil-starved condition, the bearing/retainer treatments must apparently be such that an oil coating on the bearing components of less than 0.1 μm is achieved.

Thus, in this test series, two different bearing/retainer treatments were selected which would result in an initial oil film thickness coating on the bearing components of less than 0.1 μm (4 μin.). The target values of these initial oil coating thicknesses were 0.05 μm (2 μin.) and 0.025 μm (1 μin.). It must be realized that accuracy in the treatment technique for obtaining such thin coatings is limited, and probably the target values were not precisely achieved. BBRC did use rinse-back solutions on these bearings that were estimated to achieve the desired initial oil film thicknesses. Even though the target values may not have been precisely met, the resulting coatings should have values that were approximately correct and both appeared to be less than the previously tested 0.1 μm (4 μin.) which was referred to as a "thin film" in Reference 8. Again no oil reservoirs were installed in the bearing test chamber for these short-term tests.

Unfortunately, the bearings having a target value of 0.05 μm (2 μin.) for the initial oil coating presented operational problems during testing. Therefore, after several unsuccessful attempts to test these bearings they were removed from the Test II Series schedule. The problem in testing these bearings was found to be similar in nature to the wedging between the retainers and outer rings encountered previously and discussed in detail in Reference 8. Since these bearings were employed previously in the test program, a review of the previous testing procedure revealed that the same two bearings had presented problems even when lubricated with a thick coating of BBRC 36233. Fortunately in the previous program it was possible to disassemble and reassemble the bearings and get them to operate, but after repeating this procedure it was not possible to get the bearings to operate with the very thin film of oil used in this test series. In any case, the bearings having a very thin oil coating of 0.025 μm (1 μin.) did perform satisfactorily and the film thickness results are shown in Figure 6. As normally employed, the test procedure for these tests was to run all the 22N (50 lb) load conditions, then repeat the procedure for the high load condition of 890N (200 lb). As is seen in Figure 6, at initiation of the test at the light load the measured film thicknesses for the bearings were very close in agreement with both Dowson's Eq. (3) and Grubin's Eq. (1). But, after running for five hours the dimensionless film thicknesses H and H' had

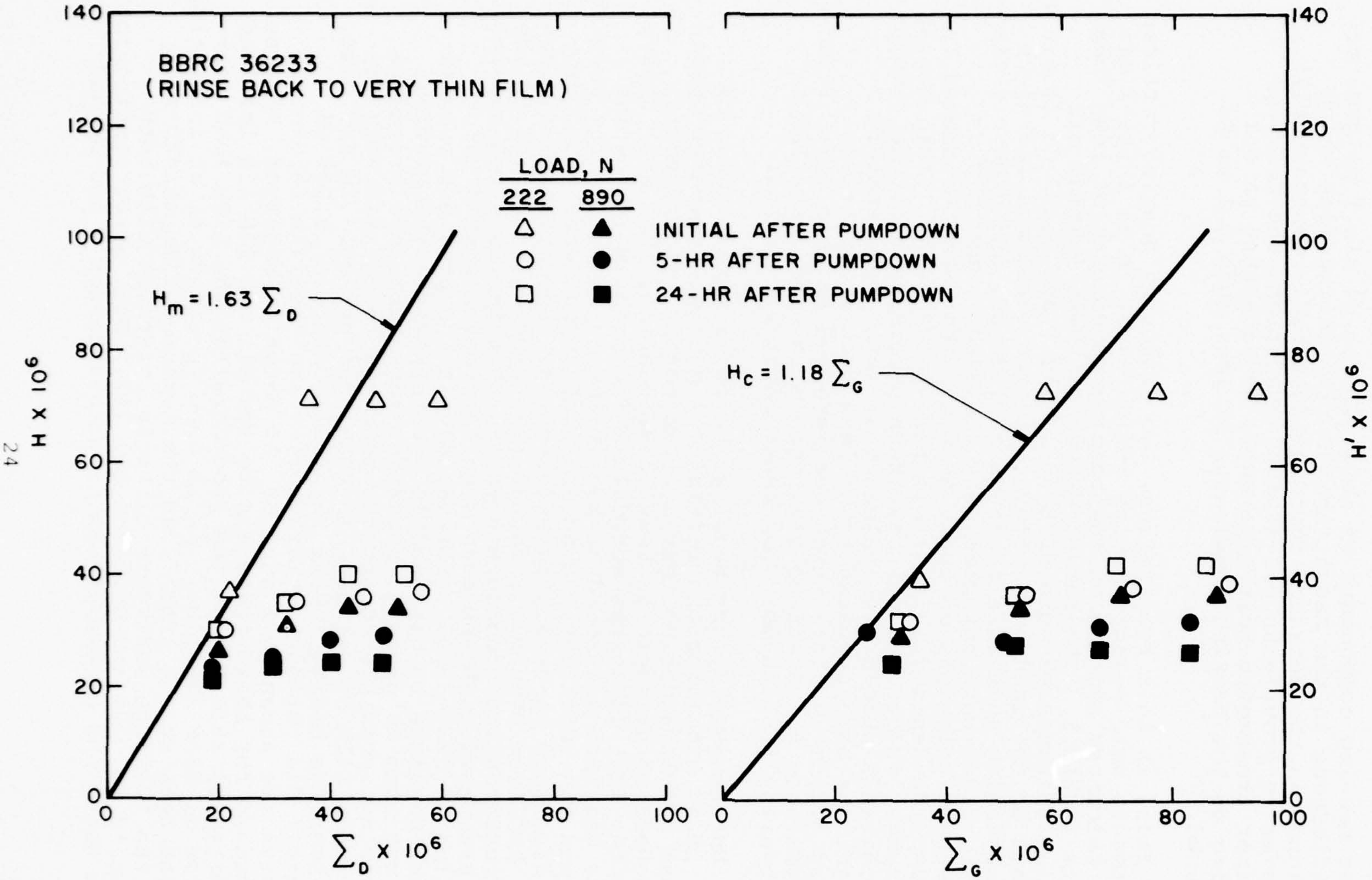


Figure 6. Dimensionless Oil Film Thicknesses for Standard Bearings Having Very Thin Coat of BBRC 36233

both leveled out at 30 x 10^{-6} to slightly over 40 x 10^{-6} for the covered range of dimensionless parameter values. For the high load test which followed, the dimensionless film thickness values decreased somewhat to values of 20 x 10^{-6} to slightly over 35 x 10^{-6}. Although this change in values may be a result of load effects which are not completely correlated by Dowson's and Grubin's parameters, it can be seen that after the initial low-load data were obtained, a limited starvation condition similar to that seen for the bearings having Nylasint 64HV-2 retainers (Fig. 5) was exhibited. Again there did not appear to be sufficient lubricant to provide flooded lubrication of the ball-race conjunctions, but examination of the test data showed there to be adequate film thickness to place operation in the EHD regime. Since the data shown in Figure 6 are for the very thin oil coat of 0.025 μm (1 μin.) in this test series, it seems logical that the somewhat thicker coat of 0.05 μm (2 μin.) would give results more nearly in agreement with Dowson's Eq. (3) and Grubin's Eq. (1) which should compare closely with data presented for bearings having a thin coat of BBRC 36233 of approximately 0.1 μm (4 μin.) in Reference 8.

Test Series III — Effect of Frequency and Extremes of Temperature Variation. The purpose of this test series was to determine the effects of the frequency and extremes of temperature variation on the formation and maintenance of elastohydrodynamic films in DMA bearings operating in vacuum. The test conditions for this series were as follows:

Bearings:	MRC standard
Bearing/retainer treatment:	Thick initial oil film
Test oils:	Apiezon A + 1.5% antioxidant + 5% lead naphthenate, and BBRC 36233
Pressure:	Equilibrium vapor pressure of test oil
Temperature:	Cycled with varied frequency and varied extremes between 24 and 66C (75 and 150F)
Load:	890N (200 lb)
Speed:	100, 200 rpm

As seen in the above list of conditions, the bearings employed in this test series were MRC standard bearings with phenolic retainers. In order to be able to compare with previous results(8) using these same bearings and oils, the bearing/retainer treatment resulted in a thick initial oil film coating (between 3 and 4 μm). Also, as seen in the above list it was scheduled to use two test oils. The bearings lubricated with low-viscosity Apiezon A plus additives oil could not initially be made to operate because of the wedging problem discussed earlier in this report. The bearings were finally made operational after several disassemblies, swapping of retainers between the two bearings and reversing the installation of the retainers. The measured ΔLy for this test, after operation of the bearings was achieved, was zero, giving no film thickness at any of the ball-race conjunctions. On the other hand, the bearings lubricated with BBRC 36233 operated very well with no wedging problem, and the data will be presented shortly.

As has been previously discussed, for moderate changes in bearing temperature, the effects of temperature variation on the EHD film thickness behavior are well correlated by dimensionless parameters which include the effects of temperature through viscosity changes. However, the question arises as to whether or not such parameters can account for the effects of wider ranges of temperature variation and the temperature cycling such as occurs in actual space hardware. For example, changes in temperature distribution among internal components of DMA systems due to environmental temperature changes may alter the molecular migration or other lubricant transfer mechanisms within the system thus affecting the residual oil film thicknesses on the bearing components. This in turn may affect the EHD film thicknesses which develop in the bearings. One question to be answered in this test series is whether or not the EHD film thickness in a bearing will completely recover from the effects of a temperature cycle. That is, will the EHD film thickness return to the same value as before the temperature excursions occurred.

Because the tests in this series might involve the effect of lubricant transfer between bearings and other components of the test chamber, lubricant reservoirs were installed in the test chamber for this test series.

The temperature cycles in these experiments were produced by installing two band heaters (900 watts each) around the outside of the test chamber at the bearing locations and controlling the power to the heaters with a variable autotransformer. The technique was found to be superior to using infrared heat lamps.

Figure 7 shows the dimensionless film thicknesses, H and H', plotted versus Dowson's and Grubin's parameters respectively for the standard bearings lubricated with BBRC 36233.

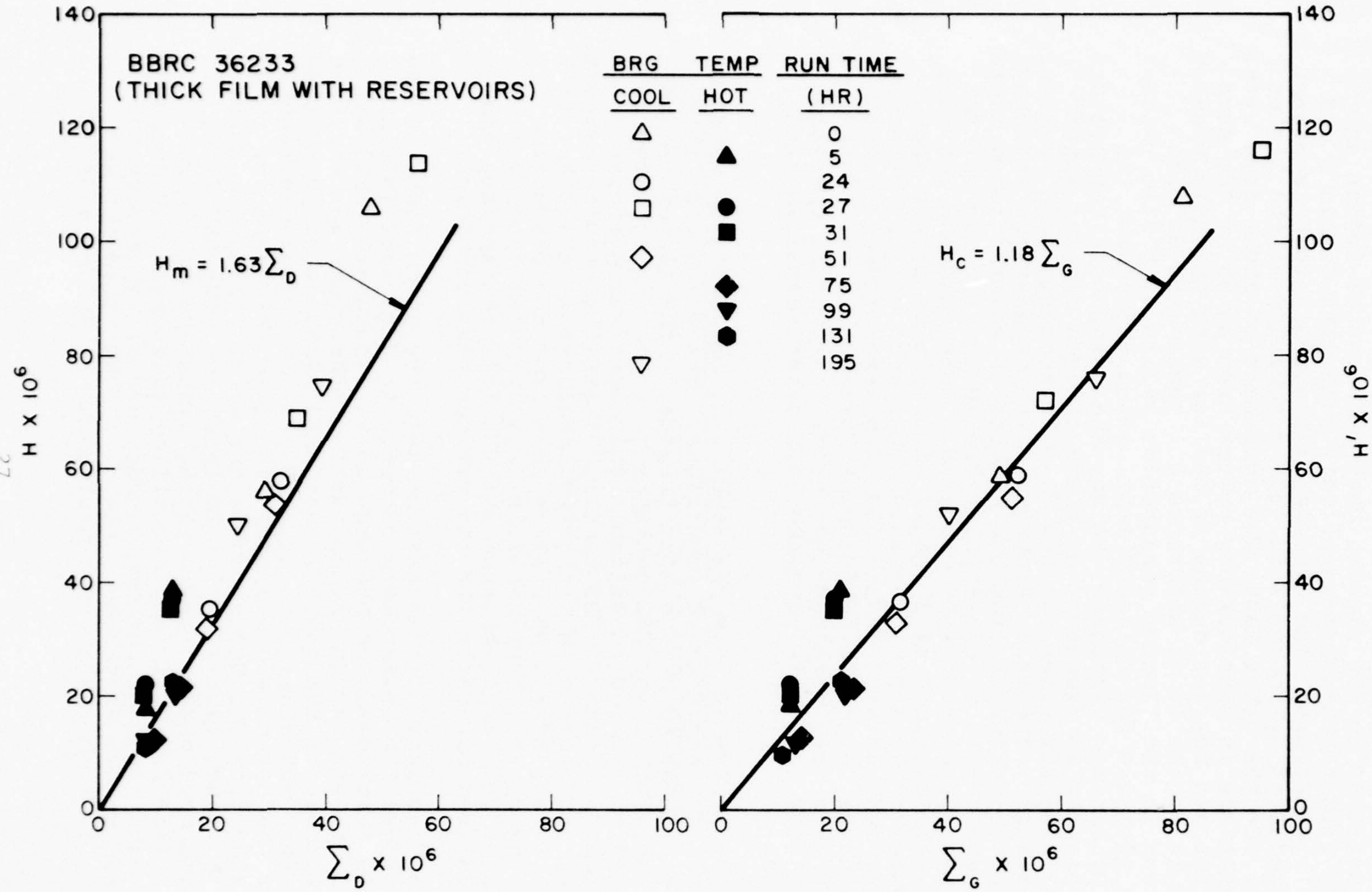


Figure 7. Dimensionless Oil Film Thicknesses for Standard Bearings Subjected to Temperature Variations While Operating at a Constant Load of 890 N

As illustrated in the figure, both the central-region and minimum film thicknesses agree very well with Grubin's and Dowson's equations. The minimum film appears to be somewhat underpredicted by Dowson's Eq. (3) especially at the higher values. On the other hand, the agreement of these data with the equations is considerably better than shown by the same lubricant and bearing design in earlier work which is presented in Figure 16 of Reference 8. These data do show operation in the flooded conjunction regime with film thicknesses adequate for full EHD lubrication throughout the range of testing variables. Also, the changes in temperature and variation in frequency of these changes did not appear to significantly alter the EHD film thicknesses developed in the bearings. The film thicknesses in the bearings appeared to completely recover from the effects of several temperature cycles and return to essentially the same values as before the temperature excursions occurred. Both cold and hot data points were obtained at 27 hours of operation as are shown in Figure 7. To accomplish this, operation of the hot bearings was stopped at 27 hours and they were allowed to cool for 69 hours before being put back into operation. Therefore, the hot data were obtained just prior to stopping operation and the cold data were obtained just after restarting the bearings.

Test Series IV — Model Studies of EHD Contacts under Fully Flooded and Starved Conditions. The purpose of this test series was to define more completely the factors which determine whether the EHD contacts in a bearing operates in a flooded or starved condition. This work was performed using the SwRI optical EHD tester,[9] and the results support the work performed with the actual bearings in the other test series of this task. As will be seen from the discussion below, the optical EHD tester is ideally suited for basic studies of this type, because the EHD film thicknesses can be visualized optically for the various test conditions.

For one part of this test series, the glass disks of the SwRI optical EHD tester were coated with different initial oil film thicknesses on one side only. The coat of oil was applied at SwRI and no "rinse back" of the disks was employed. The thickness of the oil film was determined by weighing the disk before and after coating, and assuming equal distribution of the oil on the surface. The film thickness was then calculated from the before and after weigts using the surface area. For each initial oil film thickness, the EHD films that developed between the rotating disks and a 2.54-cm (1-in.) diameter ball in pure rolling were optically measured. These tests were all performed under normal laboratory temperatures at two different loads of 17.70N (3.98 lb) and 59.78N (13.44 lb) which give maximum Hertz stresses of

41,370 N/cm^2 (60,000 $lb/in.^2$) and 62,055 N/cm^2 (90,000 $lb/in.^2$), respectively. The oil temperature at the conjunction inlet was measured and accounted for in the EHD calculations. The range of initial oil film thicknesses employed was such that operation extended from the severely starved to the fully flooded regimes. These experiments were limited to one test oil, BBRC 36233.

For presentation of data obtained from these experiments the central-region and minimum film thickness equations discussed in Reference 8 were employed. These equations for ball-on-disk experiments are repeated below for the convenience of the reader and are as follows:

For the central-region film thickness:

$$H_c = 1.05\ \Sigma_c \tag{6}$$

where $H_c = \dfrac{h_c}{R}$ = dimensionless central-region film thickness

$\Sigma_c = \dfrac{(GU_t)^{0.74}}{(W')^{0.074}}$ = dimensionless material-velocity-load parameter for central-region film thickness correlation for circular conjunctions

For the minimum film thickness:

$$H_m = 0.75\ \Sigma_m \tag{7}$$

where $H_m = \dfrac{h_m}{R}$ = dimensionless minimum film thickness

$\Sigma_m = \dfrac{G^{0.70}\ U_t^{0.77}}{(W')^{0.14}}$ = dimensionless material-velocity-load parameter for minimum film thickness correlation for circular conjunctions

W' = dimensionless load = $\dfrac{P}{\overset{*}{E}R^2}$

P = load

and all other symbols are defined after Eq. (4). The appropriate parameters from these Eqs. (6) and (7) were employed for correlation purposes and are presented in the figures that follow.

Figures 8 and 9 show similar relationships between the central-region and minimum film thicknesses for the three different levels of initial oil film thickness (oil coating) on the disks. As can be seen when comparing the two figures, the relative position of the drip-feed curve (dashed-line) is somewhat different. Although it is located between the 40.26 μm (1585 μin.) and 17.78 μm (700 μin.) curves for central-region film thicknesses and between the 17.78 μm (700 μin.) and 11.02 μm (434 μin.) curves for the minimum film thicknesses, what seems to be of more significance is the fact that the drip-feed lubrication, which was thought to supply maximum flooded conjunctions, does not appear to produce conjunction separation between the ball and disks as great as the thicker initially applied oil films. Also shown by these data is the severe starvation of the ball-disk conjunctions especially at the higher Σ_c and Σ_m values for the thinner initial oil film coating of 11.02 μm (434 μin.). The minimum film thicknesses displayed very severe starvation at Σ_m values greater than 45×10^{-6}. Tests performed at four initial oil film coatings thinner than the 11.02 μm (434 μin.) thickness test showed extremely severe starvation of the conjunctions and meaningful data could not be obtained. Therefore, the 11.02 μm (434 μin.) data were for the thinnest oil coating that could be interpreted. Also of interest is that according to information furnished by BBRC for bearings having what they considered to be a "thick film" of BBRC 36233, the lubricant would have a film thickness coating of 3 to 4 μm (118 to 157 μin.). As discussed above, ball-disk experiments having initial oil film coatings of this thickness exhibited severely starved contacts, whereas the bearings treated by BBRC did not show near this degree of starvation (Fig. 4). In fact, bearings treated with the BBRC "thin film" and having what BBRC considered to have a film thickness coating of approximately 0.1 μm (4 μin.) did not give measured conjunction film thickness data significantly different than those for the "thick film" bearings.(8)

This rather wide discrepancy between the EHD film thickness results obtained with the bearings versus those obtained in the ball-disk experiments could be at least partially explained if the residual oil film thickness on the bearing components was actually thicker than the values provided by BBRC. It should be emphasized that the residual oil film thickness on the bearing components is an estimate made by BBRC, and is based on experiments conducted by them. Since it is impractical to determine the residual oil film thickness on the bearings by weighing each bearing before and after treatment, BBRC estimates the residual oil film thickness

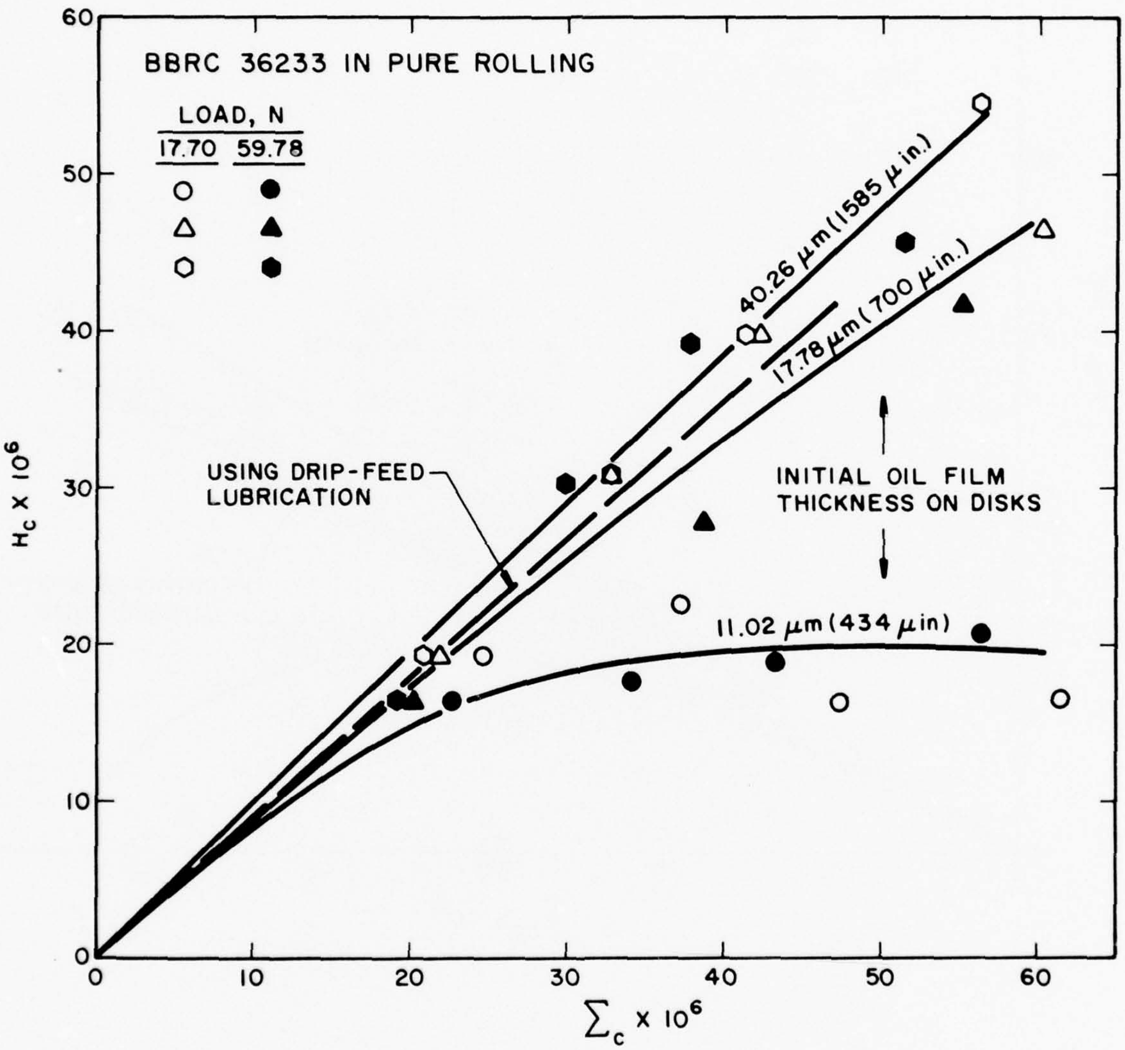


Figure 8. Dimensionless Central-Region Oil Film Thickness in Ball-Disk Conjunctions for BBRC 36233

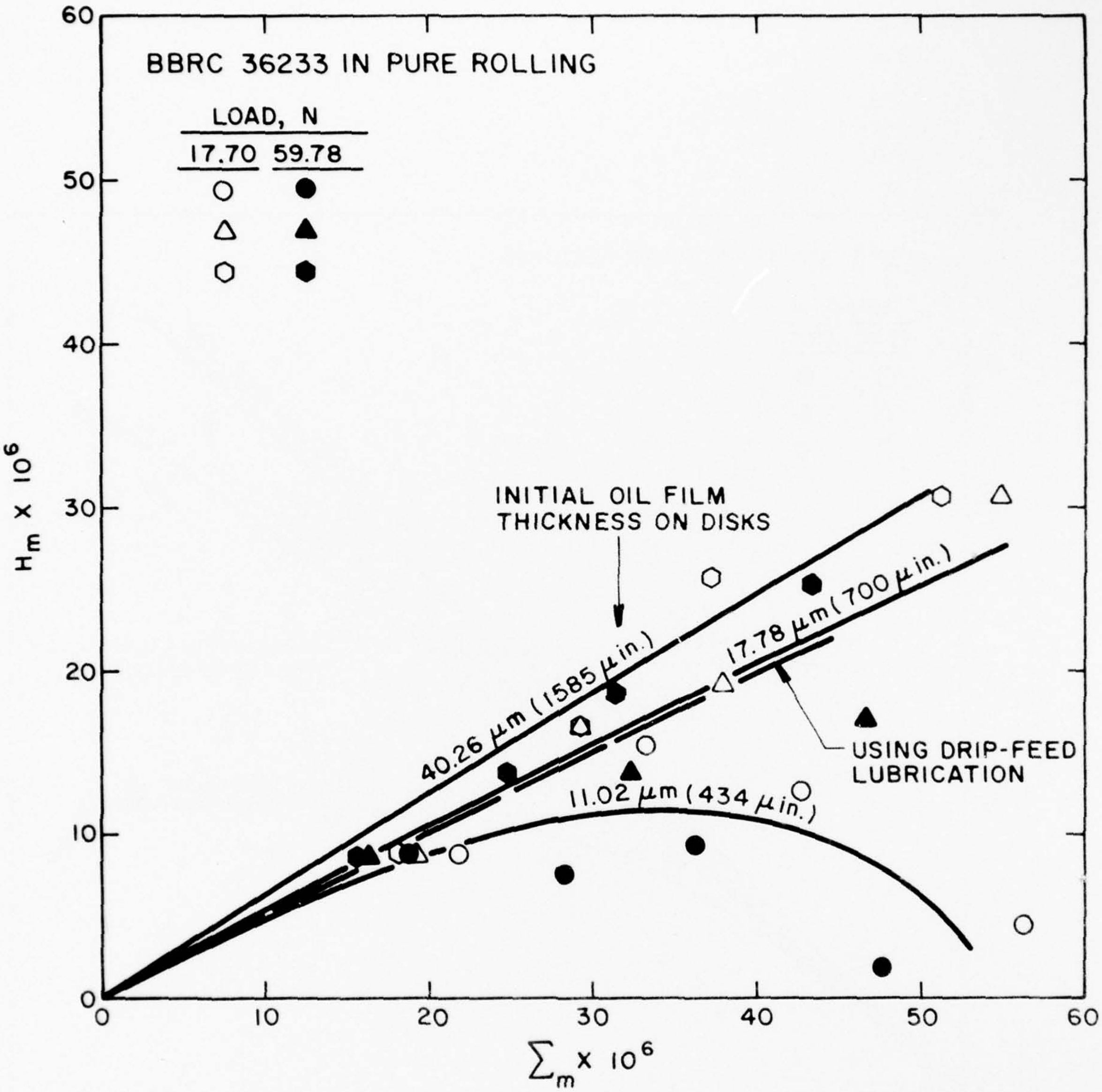


Figure 9. Dimensionless Minimum Oil Film Thickness in Ball-Disk Conjunctions for BBRC 36233

from data determined in experiments with metal samples. These metal samples were subjected to the various oil treatment processes, and the residual oil film thicknesses were determined by weighing the metal samples before and after oil treatment, as was done with the glass disks in the present work. It is then assumed that for a given oil treatment, the bearing components will exhibit the same residual oil film thickness as the metal sample. It is therefore possible that the residual oil film thickness on the bearing components will be different, and indeed thicker, than those on the metal samples. This is because oil can be retained in the bearings by surface tension effects at the interfaces between the balls, races, and retainers, which does not occur with the metal samples.

Another possible explanation for the discrepancy involves considerations of the wettability of the oil on the glass disks. It is likely that the wetting of the cerium-oxide-coated glass disks by the oil is poorer than occurs on the metallic bearing components. If this is the case, then to generate a given EHD film thickness in the ball-disk conjunction would probably require the application of a thicker oil film coating to override the influence of the poorer wettability. It should also be mentioned that the ball in the ball-disk experiments was not coated with oil, and this may have had some influence on the results. Finally, it should not be overlooked that centrifugal forces set up by the rotating disk tend to cause an outward flow of the oil on the disk surface which could tend to thin the oil coating on the disk, although this was not observed in the tests.

In another phase of this test series, the effects of retainer feed on starvation was studied. Retainer models were fabricated of Grade LBB laminated phenolic, Meldin 9000 porous polyimide, and Nylasint 64HV-2 porous nylon according to the design shown in Figure 10. These models were treated by BBRC with BBRC 36233 oil and should have been left in a saturated condition. BBRC data show that the phenolic, polyimide, and nylon models held 13, 25, and 23 percent of oil by weight respectively. With each model and a 1.5875-cm (0.625-in.) diameter ball installed in the tester as shown in Figure 11, the EHD contact was observed optically to determine oil feed characteristics of the various materials, and the resulting EHD film thicknesses. The retainer model was loaded against the ball with a mechanical force gage (Hunter Spring Model No. L-5M) as shown in Figure 11, and for these experiments the two loads employed were 1.11N (0.25 lb) and 2.22N (0.50 lb). The load between the ball and glass disks was 17.70N (3.98 lb) giving a maximum Hertz stress of 56,539 N/cm^2 (82,000 $lb/in.^2$). This was the same load used in

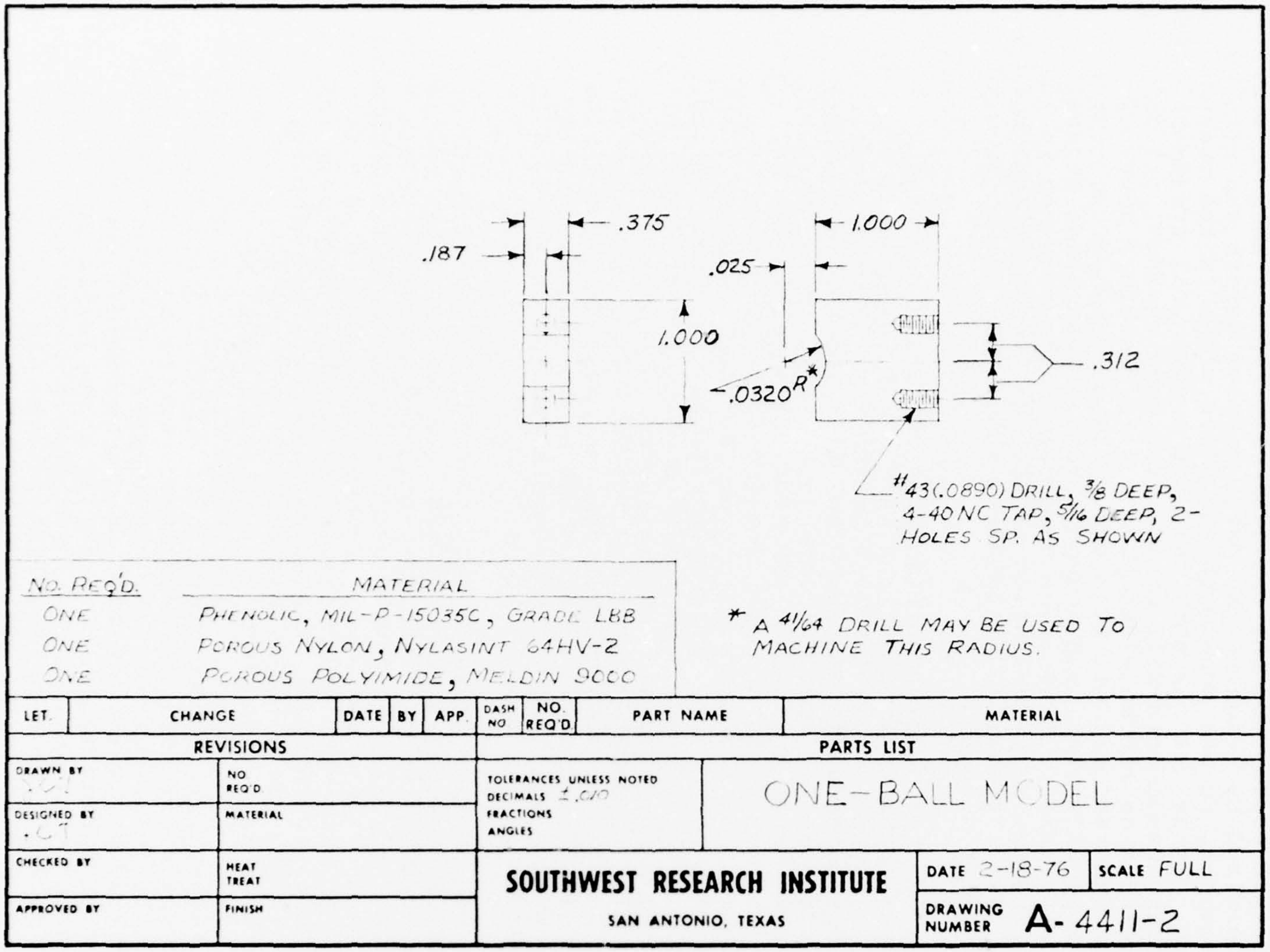


Figure 10. Design Features of One-Ball Model

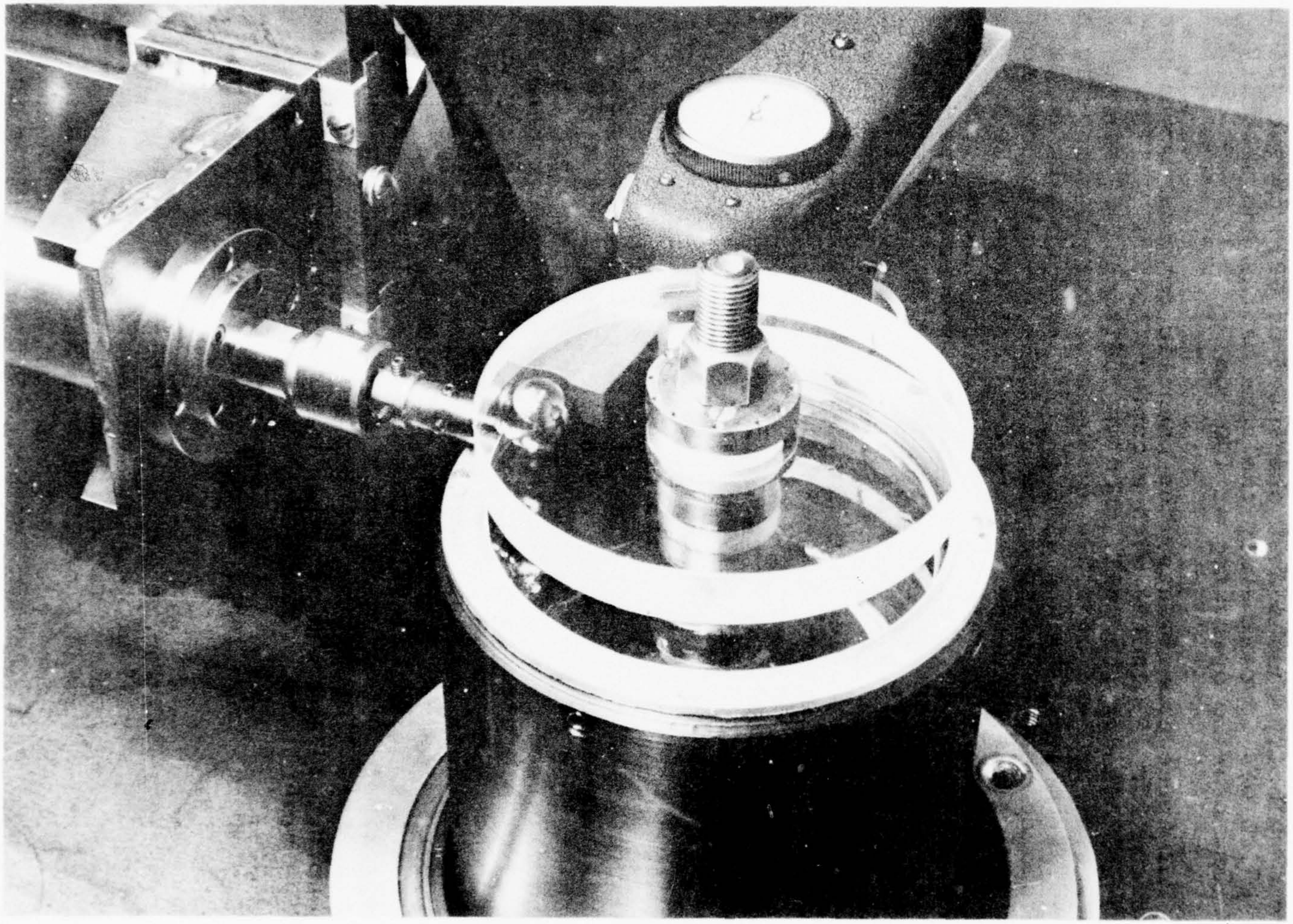


Figure 11. One-Ball Model Setup for Oil-Feed Study

previous optical rig tests[8] which gave a Hertz stress of 41,370 N/cm^2 (60,000 lb/in.^{2}) using a 2.54-cm (1-in.) diameter ball. For these experiments a smaller 1.5875-cm (0.625-in.) diameter ball was employed giving the higher Hertz stress. All of these experiments were performed in pure rolling and the sum velocity was held at one speed of 63.5 cm/sec (25 in./sec) which was approximately the same as the sum velocity of a test bearing operating at 100 rpm.

For these experiments, the phenolic retainer model was first run for 15 minutes with a contact load of 1.11N (0.25 lb) against the ball, then an additional 20 minutes with a contact load of 2.22N (0.50 lb). During this time the ball-disk conjunction was constantly viewed showing no oil feed whatsoever. This suggests that the phenolic does not feed and agrees with the bearing data obtained in Test Series I of this Task. An interesting observation after concluding this experiment was that a drop of oil placed on the exposed layered surface of the phenolic model would soak into the material, indicating that it was not fully saturated, although the data for this model as supplied by BBRC gave an oil content by weight of 13 percent. It should also be noted that 13 percent seems extremely high, especially since the bearing retainers were fabricated from the same piece of material and contained 7 percent by weight of oil. Normally, a regular phenolic material meeting MIL-P-15035C specifications will only hold 3-5 percent oil.

The porous polyimide retainer model was also run for 15 minutes with a contact load of 1.11N (0.25 lb) against the ball and an additional 20 minutes (35-min. total) with a contact load of 2.22N (0.50 lb). During this time there was no oil feed observed for this material. After this 35 minutes of operation the retainer model was completely unloaded and reloaded to 2.22N (0.50 lb) twice. After the second loading a yellow streak indicating some oil feed began to appear. The oil continued to slowly feed, and streaks in the conjunction oil film increased to a thickness value of approximately 0.1 μm (4 μin.) after another 15 minutes of (50-min. total) operation. After this the test was continued for another 25 minutes (75-min. total), but the feed appeared to have stabilized, and the observed conjunction remained unchanged. It appeared that the retainer model was feeding enough to give some lubrication, but the contact remained very much starved.

The porous nylon retainer model began to display oil feed immediately after the test was started using the 1.11N (0.25 lb) load between the ball and the retainer model. This load was held constant for 17 minutes and then increased to 2.22N (0.50 lb) where the test was continued for an additional 26 minutes (43-min. total). During this time the oil feed from the retainer model continued, and streaks in the conjunction oil film increased to a thickness of approximately 0.2 μm (8 μin.). After 43 minutes of total run time the conjunction was viewed at retainer loads of 0, 1.11, and 2.22N (0.25, and 0.50 lb) and the following observations were noted:

a. Under loaded conditions it appeared that a center strip of the ball-disk conjunction was being wiped nearly clean and oil was squeezed from the retainer model. This oil was carried into the side areas of the conjunction where a film was formed and in fact side lobes(9) were visible.

b. When the retainer model load was increased, the thin center strip became wider. On the other hand, when the load was decreased to essentially zero the film was thin, but fairly evenly distributed throughout the contact.

c. In review, it appeared necessary to have a load to force the oil from the retainer model, but too much load between retainer model and ball would tend to wipe the ball track clean.

In conclusion, for these retainer model experiments it can be said that the phenolic displayed no oil feed while the porous polyimide had limited oil feed, and the porous nylon demonstrated the best oil feed of these three materials. In general these results tend to agree with the Test Series I bearing results, except for the bearings having polyimide retainers, where there seemed to be no oil feed present.

In the third and final phase of this test series, a study of the influence of design and operating variables, namely ball spacing, on starvation was made. For this phase, it was planned to fabricate a multi-ball model from a single material which would accommodate two or more balls at various spacings between the balls. But, actually three multi-ball models were fabricated from three different materials according to the design details shown in Figure 12. Notice the similarity between this design and the bearing retainer design shown in Figure 2. The three materials employed were Meldin 9000 porous polyimide, Nylasint 64HV-2 porous nylon, and Nylasint M-4 low-porosity nylon, all fully impregnated with 24, 25, and 2 percent by weight of BBRC 36233 oil, respectively. Since the two more porous materials, Meldin 9000 and Nylasint 64HV-2, were not needed for the ball-spacing experiments, they were employed for additional oil-feed studies.

The multi-ball retainers were installed in the optical rig as shown in Figure 13 except the four balls were equally spaced around the retainers. The total weight of the retainer

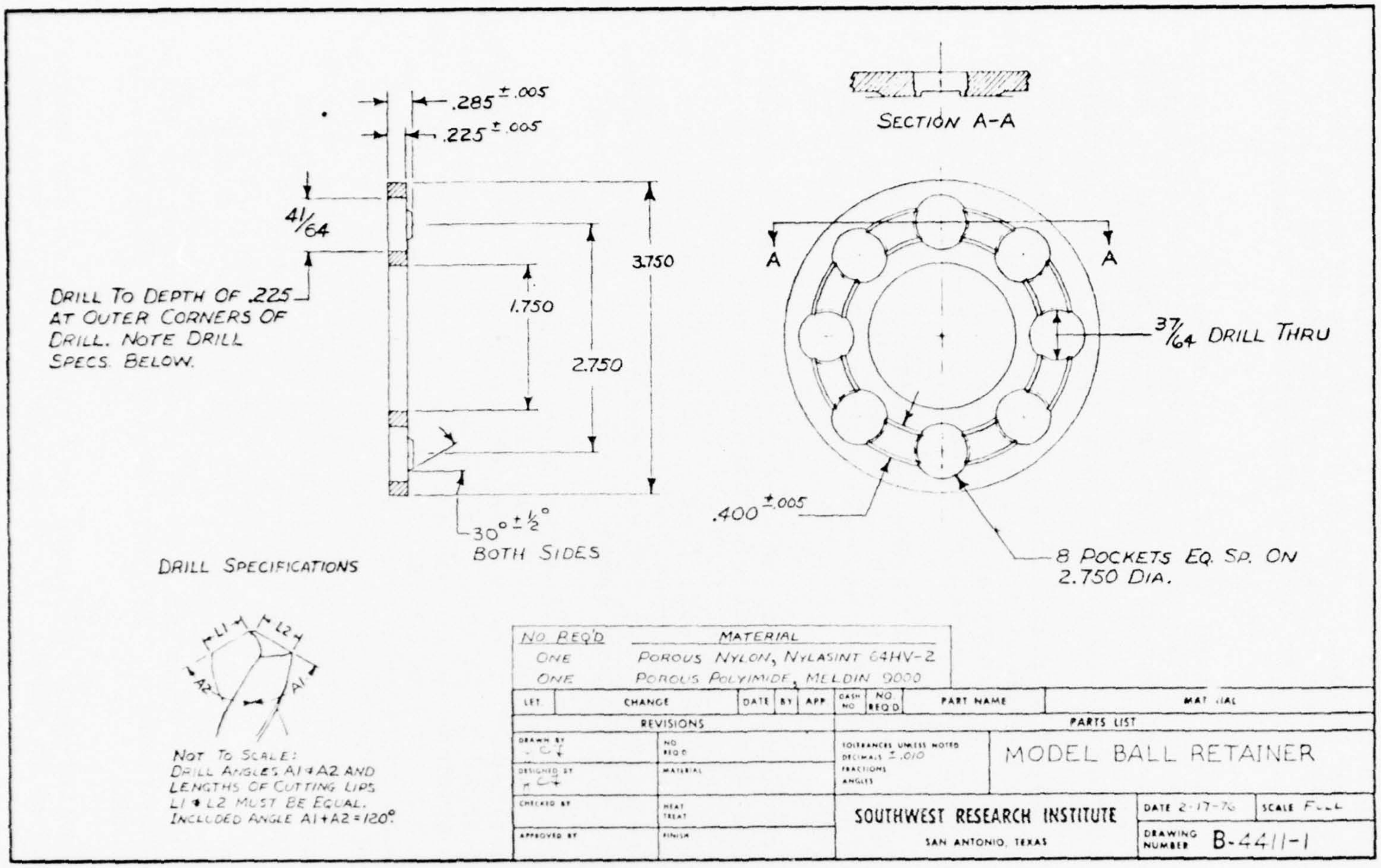


Figure 12. Design Features of Multi-Ball Model

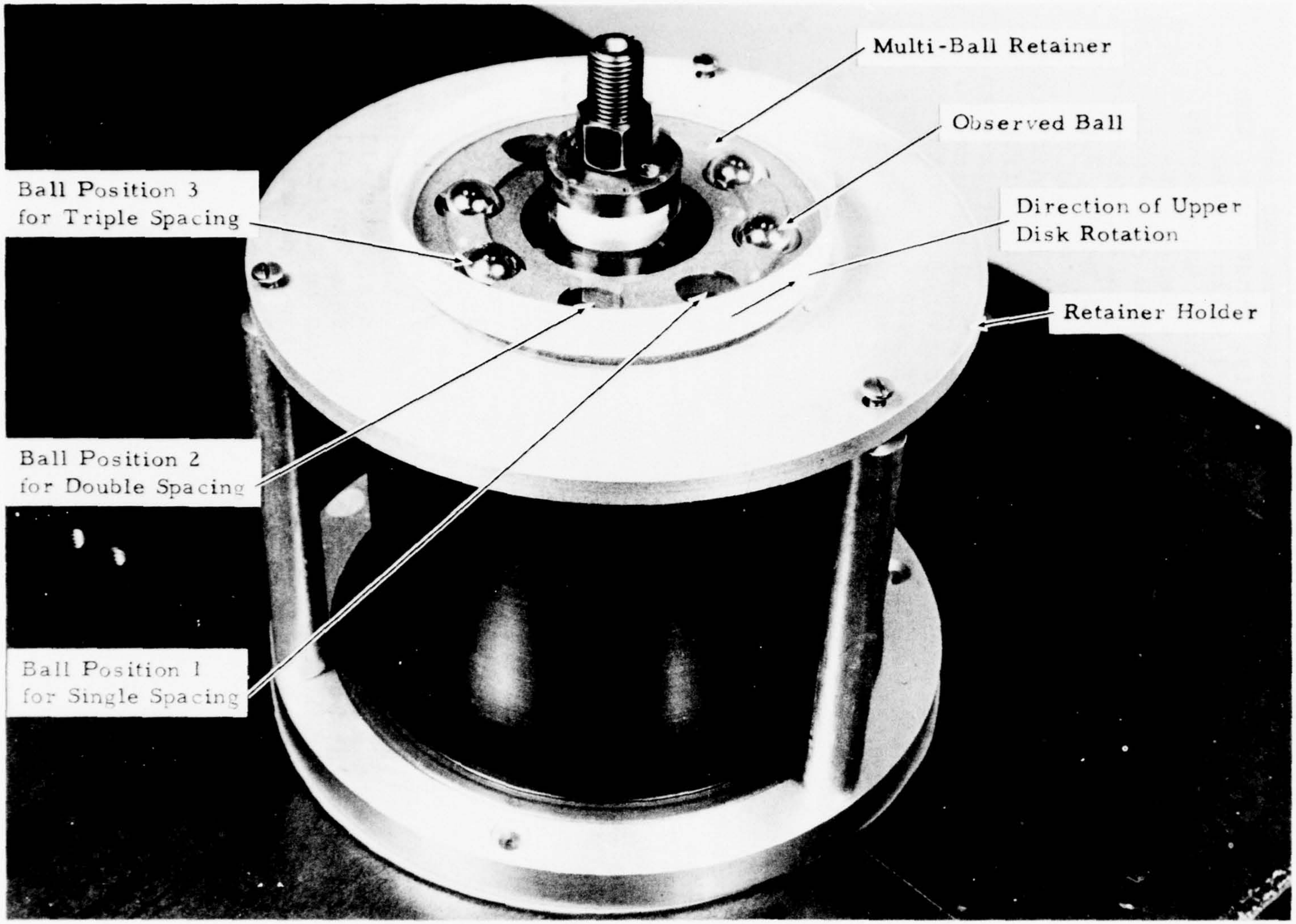


Figure 13. Multi-Ball Model Setup for Oil Starvation Studies

models and retainer holder was 3.07N (0.69 lb). Since the tabs on the retainers, as seen in Figure 12, have small cross-sectional areas and are limited in strength they were not loaded beyond this amount. Each ball pocket had two tabs which rubbed against the supporting balls. Therefore, the average load transmitted to each of the 4 balls by the retainer tabs was 0.76N (0.17 lb). For these experiments, the total load between the upper disk and 4 balls was 53.38N (12 lb), whereas the bottom disk carried a slightly heavier load due to the added weight of the balls, retainer, and retainer holder. The Nylasint 64HV-2 and Meldin 9000 were run separately under these conditions for 45 and 60 minutes, respectively, viewing alternately the four different ball-disk contacts without any evidence of oil feed from either retainer model. From these observations, it is seen that the oil feed results from these multi-ball retainer models do not agree with those results obtained from the one-ball models or the Test Series I bearing-retainer studies. Since the design of these retainers was somewhat different than the one-ball models or the standard bearing retainer design it might be expected that the oil feed results would not be the same as already discussed for these materials. On the other hand, any oil feed from the Nylasint 64HV-2 multi-ball model would have certainly supported the previously discussed experiments. Further experiments using the multi-ball retainer models were not performed since they were not originally intended for oil-feed studies.

The Nylasint M-4 low-porosity nylon retainer was employed for the ball spacing experiments. Four balls were used in this multi-ball retainer with one ball arrangement as shown in Figure 13. For this arrangement, the "movable ball" is shown in the retainer pocket at ball position 3 which gives triple spacing relative to the fixed observed ball. The two other "dummy balls" also remained in the retainer pockets as shown for all testing. For double spacing tests the movable ball shown in ball position 3 was moved to ball position 2, and likewise for single spacing the movable ball was moved to ball position 1. For all these experiments total load between the disks and balls was 53.38N (12 lb). During viewing the various ball-disk conjunctions, it appeared that at any given time one ball was not carrying any of the load. In other words, three balls were normally carrying all the load. Therefore, the maximum load on any particular ball during operation was assumed to be 17.79N (4 lb), which is 1/3 of the total load, giving a maximum Hertz stress of 56,339 N/cm^2 (81,710 $lb/in.^2$). Lubrication was provided by applying four drops of BBRC 36233 oil to the top of each ball, for a total of 0.3 cc, prior to installing the upper disk and bringing it in contact with the balls. This was done with eight balls (full complement) in the multi-ball retainer, and the rig was run for 30 minutes. Then four of the balls were removed from the retainer, and all of the experiments were performed without further lubrication. The experiments were carried out in pure rolling between balls and disks with sum velocities of 25.4, 38.1, and 50.8 cm/sec (10, 15, and 20 in./sec), and at a laboratory temperature of 25 C (77 F). This range approximates the sum velocity range encountered in the test bearings.

Figure 14 shows a comparison of the experimentally-determined dimensionless film thicknesses H_m and H_c for three different ball spacings as compared to the standard drip-feed lubrication system. As can be seen, there is very good agreement between all the data, thus negating the idea, at least for these experiments, that the ball ahead of the one under observation tends to push the oil out of the running track on the glass disks and thus restrict the oil supply to the ball being observed. In fact, at the lower material-velocity-load parameter, Σ_m, the single ball spacing gave a minimum film thickness almost twice that for both the double- and triple-spacing data. This seems the reverse of what would be expected unless the balls are actually carrying the oil into the conjunctions. On the other hand, the central-region film thickness data did not show any variation for the different ball spacings.

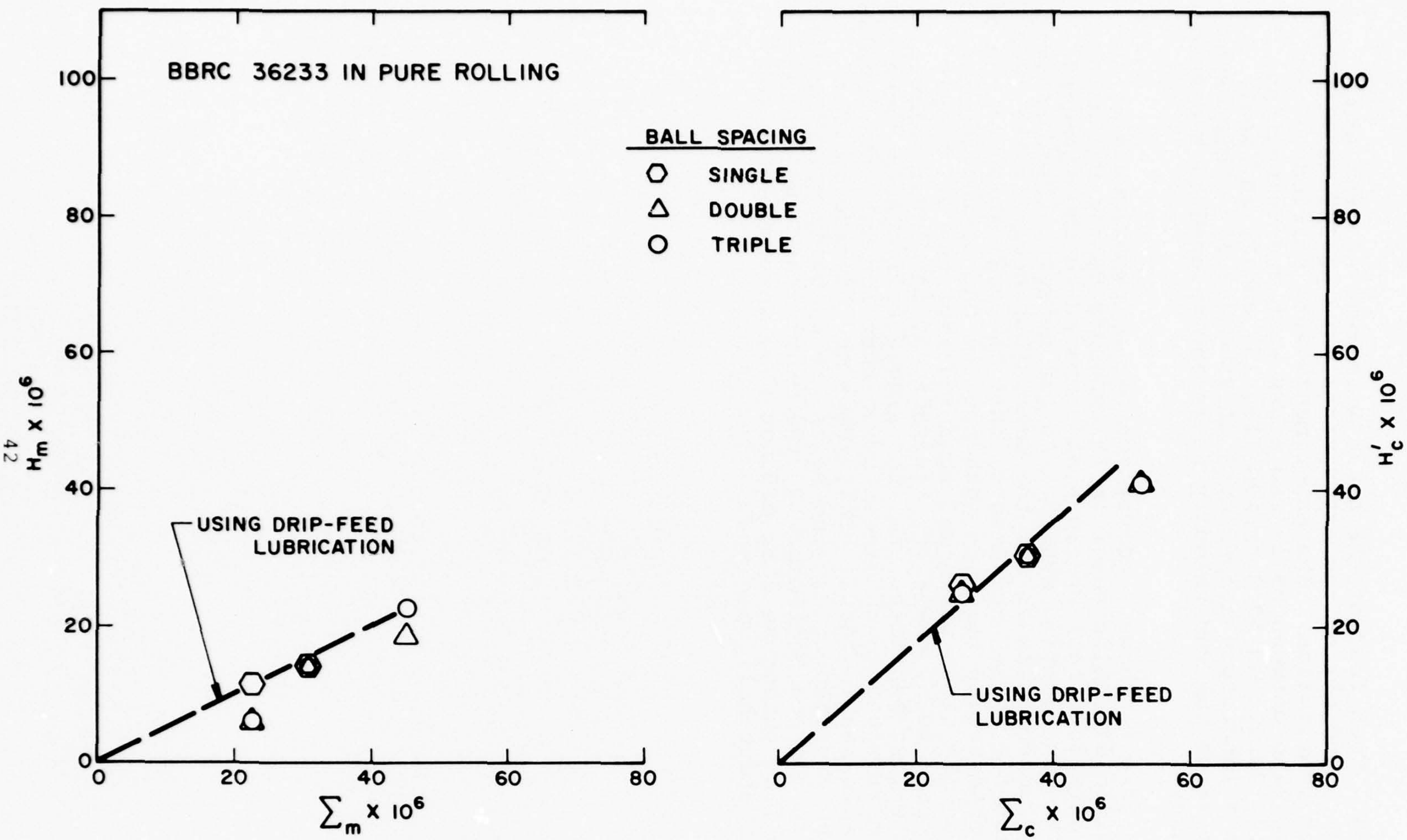


Figure 14. Dimensionless Oil Film Thicknesses for Varied Ball Spacing in Multi-Ball Retainer

SECTION V

TASK III — INVESTIGATION OF THE INFLUENCE OF LUBRICANT FILM THICKNESS TO BALL-RACE COMPOSITE SURFACE ROUGHNESS RATIO ON BEARING-LUBRICANT PERFORMANCE AND LIFE EXPECTANCY IN A SPACE (VACUUM) ENVIRONMENT

1. General

The main purpose of this task was to investigate the influence of the ratio, Λ_m, of the EHD oil film thickness to ball-race composite surface roughness on bearing-lubricant performance and life expectancy as typified by satellite stabilization methods in actual service. The overall objective of this task was to obtain data to provide a foundation for the development of accelerated tests which can be used to predict bearing-lubricant life expectancy in actual space flight service. As has been previously discussed,[8] it is believed that the most realistic way to conduct such accelerated tests is through control of the Λ_m ratio. Accordingly, the Task III tests to be discussed below were designed to provide data on bearing operation at different levels of the Λ_m ratio.

Before discussing the proposed Task III tests in detail, some general comments concerning the selection of test variables should be made. First of all, there were numerous possible test variables which deserved study. These included such parameters as bearing retainer processing variations, inlet contact starvation phenomena, the ability of contemporary/projected retainer materials to provide satisfactory long term lubricant supply, and additive and additive concentration variations.

In order to make the selection of experimental variables to be studied, the results of the work conducted during the previous two years, as reported in References 7 and 8, provided valuable guidance. Firstly, definite problems with bearing retainers were encountered during this earlier work. These problems exemplified themselves by erratic torque behavior and actual bearing failure by retainer seizure. While this could be a problem of retainer lubricant feed, which might be alleviated by use of a different retainer material, it is believed that the problem was one of retainer design and could be solved without introducing an additional variable of a different retainer material. Secondly, it seems fairly well established from the work of the previous two years that in the long-term tests, full EHD lubrication prevailed most if not all of the time. Since the range of Λ_m in these long-term tests was about 0.71 to 1.43, it was concluded that deep boundary lubrication can only be achieved at Λ_m levels somewhere below this range.

Based upon these findings, the Task III tests as outlined below were designed to achieve operation at much lower levels of Λ_m , which was the primary variable to be studied, and also to isolate the retainer problem. It was believed that this should be done with a minimum of introduction of new test variables in order that comparisons with previous results can be made. Consequently, some parameters such as bearing retainer processing variations and projected retainer materials were performed in Task II rather than in Task III. With these considerations in mind, Task III will now be discussed.

Task III consisted of three long-term tests of six months of continuous operation for each test. Each test was conducted in a separate test chamber described earlier, with two test bearings loaded against each other with a 890-N (200-lb) axial load. Bearing speed for all three tests was maintained constant at 100 rpm, and the laboratory temperature was kept approximately at 25 C (77 F).

The three separate test chambers were connected to the 1200 ℓ/sec vacuum pump, described in detail earlier,(8) and the bearing chamber pressure was the equilibrium vapor pressure of the oils. Each test chamber was equipped with an LVDT system for measurement of bearing film thicknesses, and with the necessary instrumentation for torque measurement. Both measurement systems have been discussed earlier.(7,8) Bearing film thicknesses as well as torque were measured at regular intervals throughout the entire duration of the tests.

A summary of these three tests is presented in Table 1. As shown in the table, these tests were designed to be conducted at two different Λ_m ratios, designated simply as low and medium, since at the time the tests were designed it was not known what the actual values would be. The two Λ_m levels were to be obtained by using oils of two different viscosities. The low Λ_m level would be obtained by using Apiezon A oil which has a viscosity of approximately 54 x 10^{-6} m^2/s (54 cs) at 25 C (77 F) as determined at SwRI. Within experimental error, the base Apiezon A and formulated Apiezon A oils both have this same viscosity at 25 C (77 F). The medium Λ_m level would be obtained by using BBRC 36233 oil which has a viscosity of approximately 225 x 10^{-6} m^2/s (225 cs) at 25 C.

The bearing race roughness in the transverse direction (across grinding marks), as supplied by MRC was 0.406 μm (16 μin.). This roughness is twice that of the "rough"

TABLE 1. SUMMARY OF TASK III TESTS

Test No.	Variables Studied	Oil Base Stock	Additive Package	Method of Oil Supply
1	Low Λ with high additive concentration	Apiezon A	BBRC std. (1.5% antioxidant + 5% lead naphthenate)	Thin initial film with reservoirs
2	Low Λ with low additive concentration	Apiezon A	1.5% antioxidant + 0.5% lead naphthenate	Thin initial film with reservoirs
3	Medium Λ with high additive concentration	Apiezon C	BBRC std.	Thick initial film with reservoirs

TEST CONDITIONS:

Pressure = equilibrium vapor pressure of test oils
Temperature ≈ 25 C (77 F)
Load = 890 N (200 lb)
Speed = 100 rpm

BEARINGS:

Ball-piloted phenolic retainers
Race roughness = 0.406 μm (16 μin.)

bearings employed in the earlier work reported in Reference 8. This rougher race finish was chosen for the long-term tests of this study because with the Apiezon A and C oils employed in similar tests conducted in the work cited above, operation in the boundary regime was apparently not achieved as there was no indication of wear on the balls and races of the test bearings. The objective for the long-term tests in this study was to obtain levels of Λ_m low enough to hopefully place operation in the boundary regime so that wear and possible failure due to wear would occur. Using this value of 0.406 μm (16 μin.) for the race finishes in the transverse direction and ball surface finishes of 0.025 μm (1 μin.), all provided by MRC, and employing the expression for composite surface roughness, δ_c, given in Reference 7, a value of approximately 0.33 μm (13 μin.) is obtained for δ_c. This assumes the bearing race surface finishes in the direction of grinding marks is one-half that in the transverse direction. Then using experimental values of film thicknesses as calculated from ΔLy for the three tests shown in Table 1, and employing the following equation:

$$\Lambda_m = \frac{h}{\delta_c}$$

where Λ_m = dimensionless minimum oil film thickness ratio

h = oil film thickness calculated using Eq. (38), Appendix A, Reference 8

δ_c = composite surface roughness of two bearing surfaces

values of 0.21, 0.26, and 0.78 are obtained for Λ_m for Apiezon A with high additive concentration, Apiezon A with low additive concentration, and BBRC 36233, respectively. Although, the two endurance tests using Apiezon A were designed to have the same Λ_m value, there was approximately 20 percent difference between the calculated results of 0.21 and 0.26, the lower concentration of lead naphthenate antiwear additive giving the higher Λ_m. The film thicknesses used in calculating the above Λ_m values were the average of the last ten inner-aft bearing film thicknesses that were obtained in the three long-duration tests, and these data will be included in those presented in the next subsection of this report under experimental results.

2. Experimental Results

The computer program that was developed and employed to compute the film thicknesses for these Task III tests was discussed and presented earlier.[8] Also a listing was presented in the earlier reference, but the tables of computed data for the long-duration bearing tests performed during this program are presented in the Appendix. The measured variables for a long-duration test from the previous study[8] which was extended to 12,768 hours and had a high Λ_m of approximately 1.43 is presented in Figure 15, and can be used for comparison purposes.

Figures 16, 17, and 18 show the measured variables for the three long-duration Task III tests as a function of time for this program. The film thickness data points are values that were calculated from the measured displacements, ΔLy. For these Task III tests the aft bearing-inner race contact data were plotted rather than the aft bearing-outer race contact data that were selected for plotting in the Task II tests. This was done because these inner race contact data are printed out in the Task III tables (Appendix) whereas only the equations containing the geometric constant are given in these tables for computing the outer contact film thickness values. Again as explained in subsection 2, the four different conjunction films for each test condition at any particular time are calculated from a single displacement measurement, ΔLy. The solid lines shown on the plots of H' and H versus time in Figures 16, 17, and 18 are the values that would be obtained using Grubin's Eq. (1) and Dowson's Eq. (3) for calculating central-region and minimum dimensionless film thicknesses, respectively, at the measured bearing temperatures. When comparing the data calculated from measured ΔLy values with these empirical values from the equations, it is seen that the actual film thicknesses in the bearings, for the different Λ_m ratios, are always less than both the empirical minimum and central-region values predicted by the equations. This in general agrees with the similar Task III data obtained earlier and shown in Figures 23, 24, and 25 of Reference 8. As before, this suggests less than flooded lubrication in the ball-race conjunctions. The endurance tests performed during this program gave considerably thinner film thicknesses in the bearings than those previously tested using similar oils. This implies that the rougher bearing race surfaces used in this program, which gives lower Λ_m values because of larger δ_c's, causes the thinner oil films at the ball-race conjunctions. On the other hand, the earlier short-term tests[8] do not support this observation showing essentially no difference between the film thicknesses for bearings having race roughnesses of both 0.102 μm (4 μin.) and 0.204 μm (8 μin.). Also, the earlier long-duration test having the lowest Λ_m value (0.71) and using Apiezon A with a high concentration of antiwear additive displayed film thicknesses approximately one-half those predicted by Dowson's Eq. (3) at the beginning of the test. But these film thicknesses gradually increased until agreeing reasonably well with Dowson's equation at approximately 2,500 hours of testing, and remained in fair agreement throughout the remainder of the test. In the present program, the test having the lowest Λ_m of 0.21 (Apiezon A with high concentration of antiwear additive) displayed very small film thicknesses throughout the entire 4,440 hours. Although the two other tests having Λ_m values of 0.26 and 0.78 (Apiezon A with low concentration of antiwear additive and BBRC 36233) had somewhat larger film thicknesses, they all remained significantly below either Dowson's or Grubin's predicted minimum or central-region thicknesses. In other words, data obtained during this program using the rougher bearings do appear to give thinner film thicknesses, which is somewhat contradictory to the earlier work. What is probably of more importance and will have additional discussion later in the report, is the fact that there was no evidence of rubbing, wear, or even contact between the balls and races for the extended tests using the very rough races and the low viscosity oil.

Vacuum in the chamber appeared to behave as would be expected, with the pressure first increasing when the test rigs were initially put into motion and then continuing a gradual decreasing trend until the end of testing. Between 450 and 1,600 hours the pressure ion gage was being repaired, therefore the chamber pressure was estimated as shown by the dashed line on the data plots. The measured torque in the test bearings seemed to behave fairly normal with fluctuations that appear to be reasonable and which remained consistent throughout testing. As seen from the plots, torque for Apiezon A with the low concentration of antiwear additive was in general lower than the torque for the other tests.

All three of the endurance tests completed the 4,440 hours without any bearing seizure problems which probably can be attributed to the new ball-piloted retainer design. With this design the retainer could not be pushed against the inner land of the bearing outer race and caused to wedge as was experienced in the earlier work.[8] Therefore, the problem of bearing "lockup" appears to have been remedied. The reason for the test rig to stop and restart itself as shown in Figure 18 is not know. It is thought that the motor controller failed to operate for a short period of time.

Post-test Inspection of Task III Bearings. Figure 19 shows a close-up view of the ball-piloted retainer removed from the aft bearing after Test No. 1 (Table 1) was completed. There was excessive buildup of debris on the ball-pocket tabs of the retainer as shown in the photograph. This buildup was evident on approximately one-half (180°) of the tabs while the others did not show any buildup. There

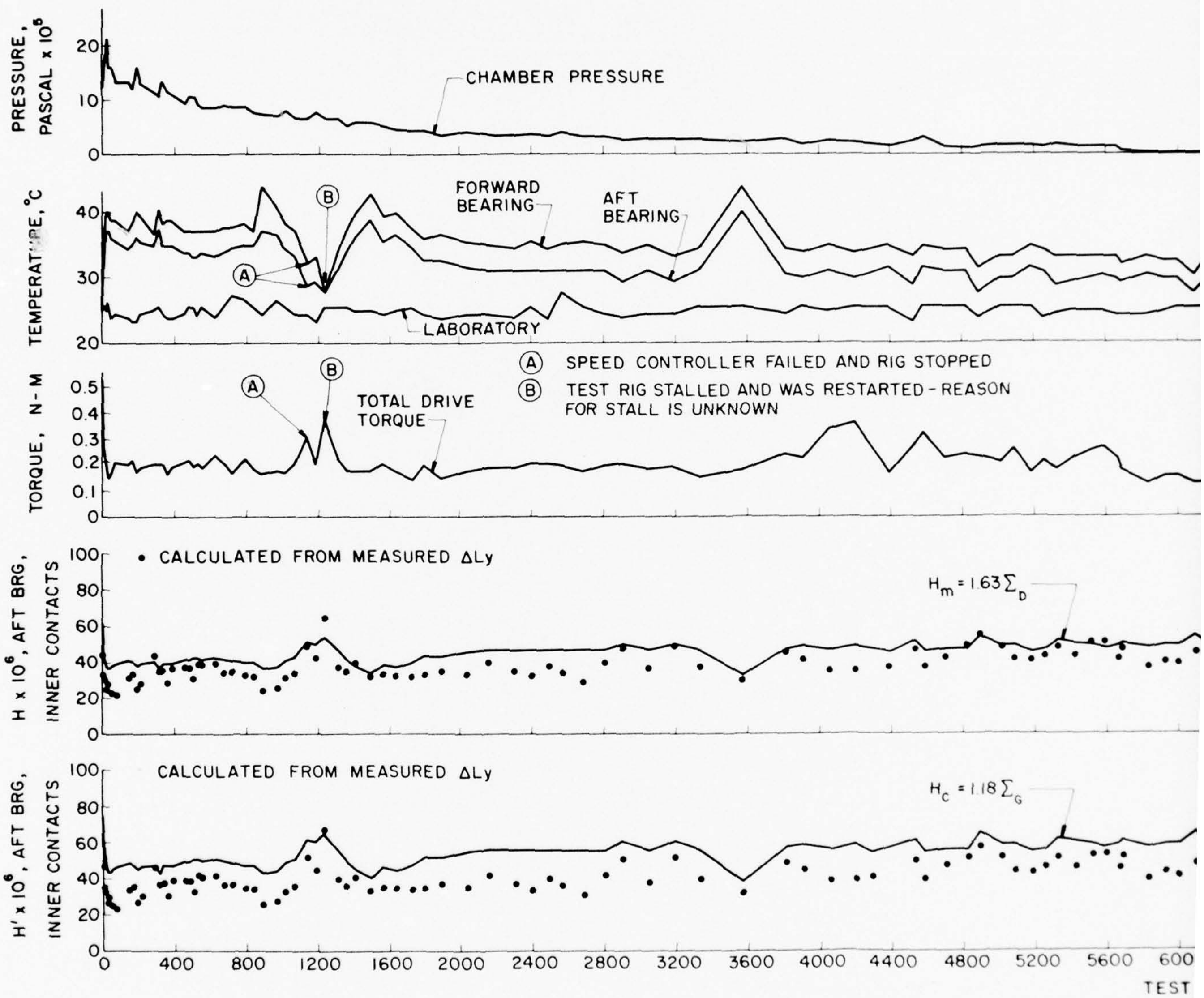
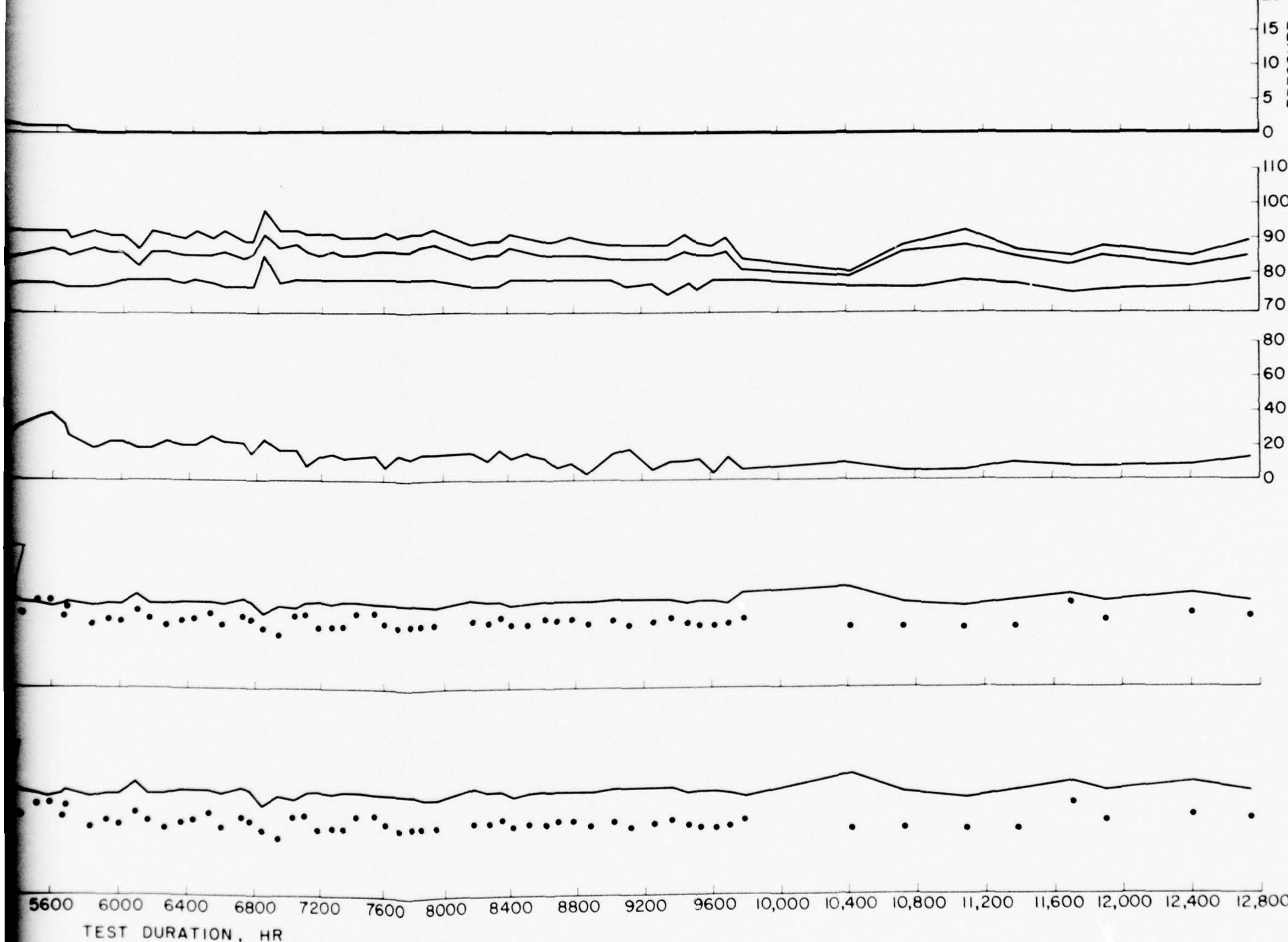


Figure 15. Measured Variables f
Lubricated With Thick Initial



riables for Endurance Test Using DMA Bearings
k Initial Oil Film of BBRC 36233, $\Lambda_m \approx 1.43$

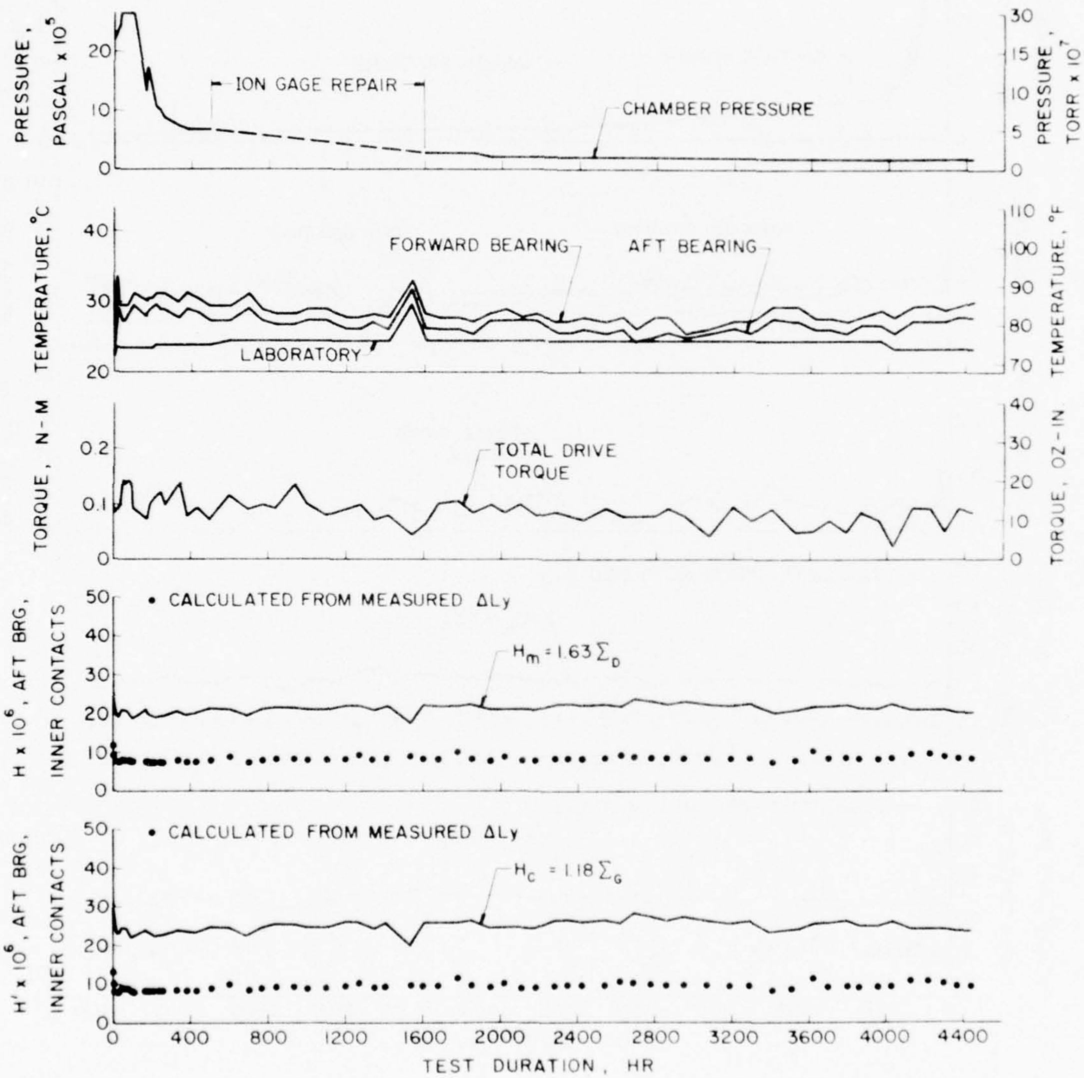


Figure 16. Measured Variables for Endurance Test Using Bearings Lubricated With Thin Initial Oil Film of Apiezon A With High Concentration of Antiwear Additive and Having Ball-Piloted Retainers, $\Lambda_m = 0.21$

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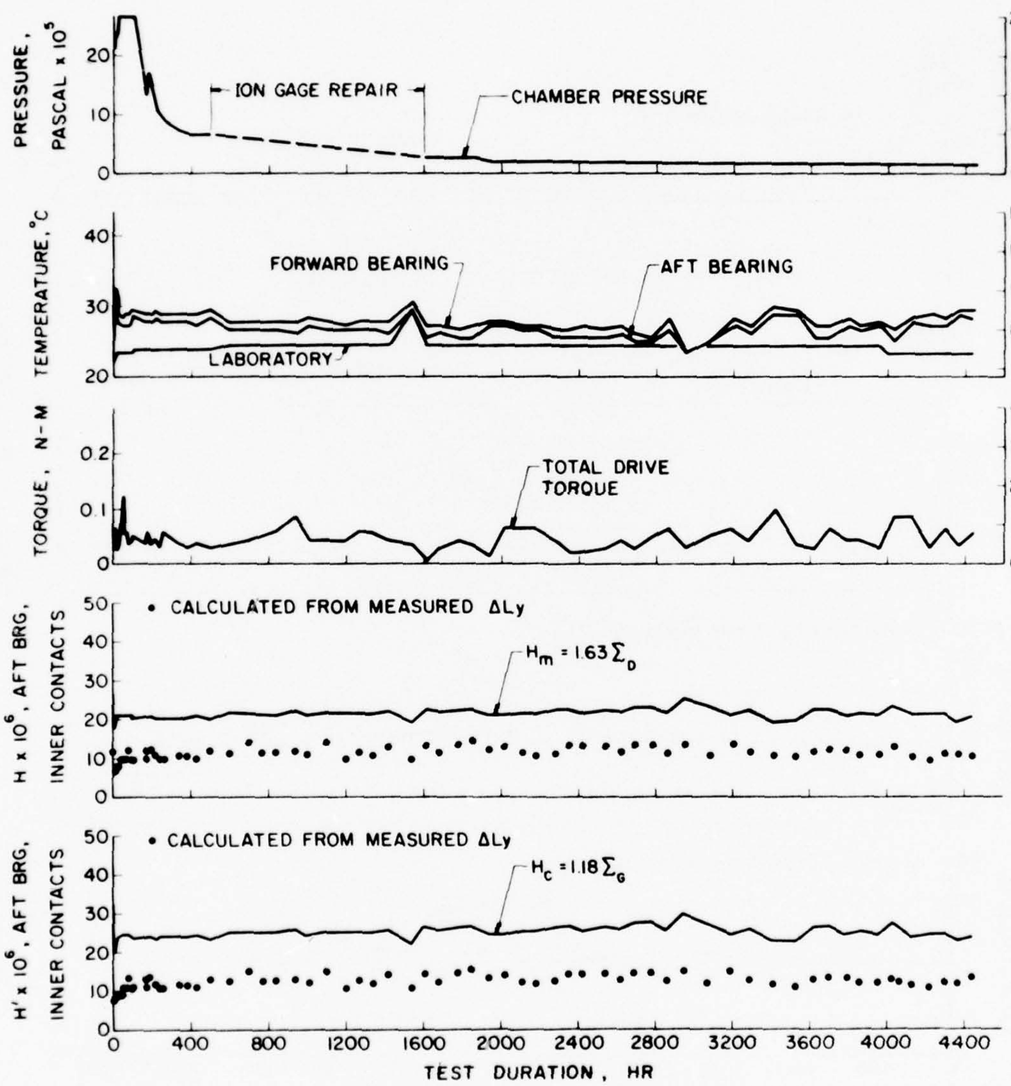

Figure 17. Measured Variables for Endurance Test Using Bearings Lubricated With Thin Initial Oil Film of Apiezon A With Low Concentration of Antiwear Additive and Having Ball-Piloted Retainers, $\Lambda_m \approx 0.26$

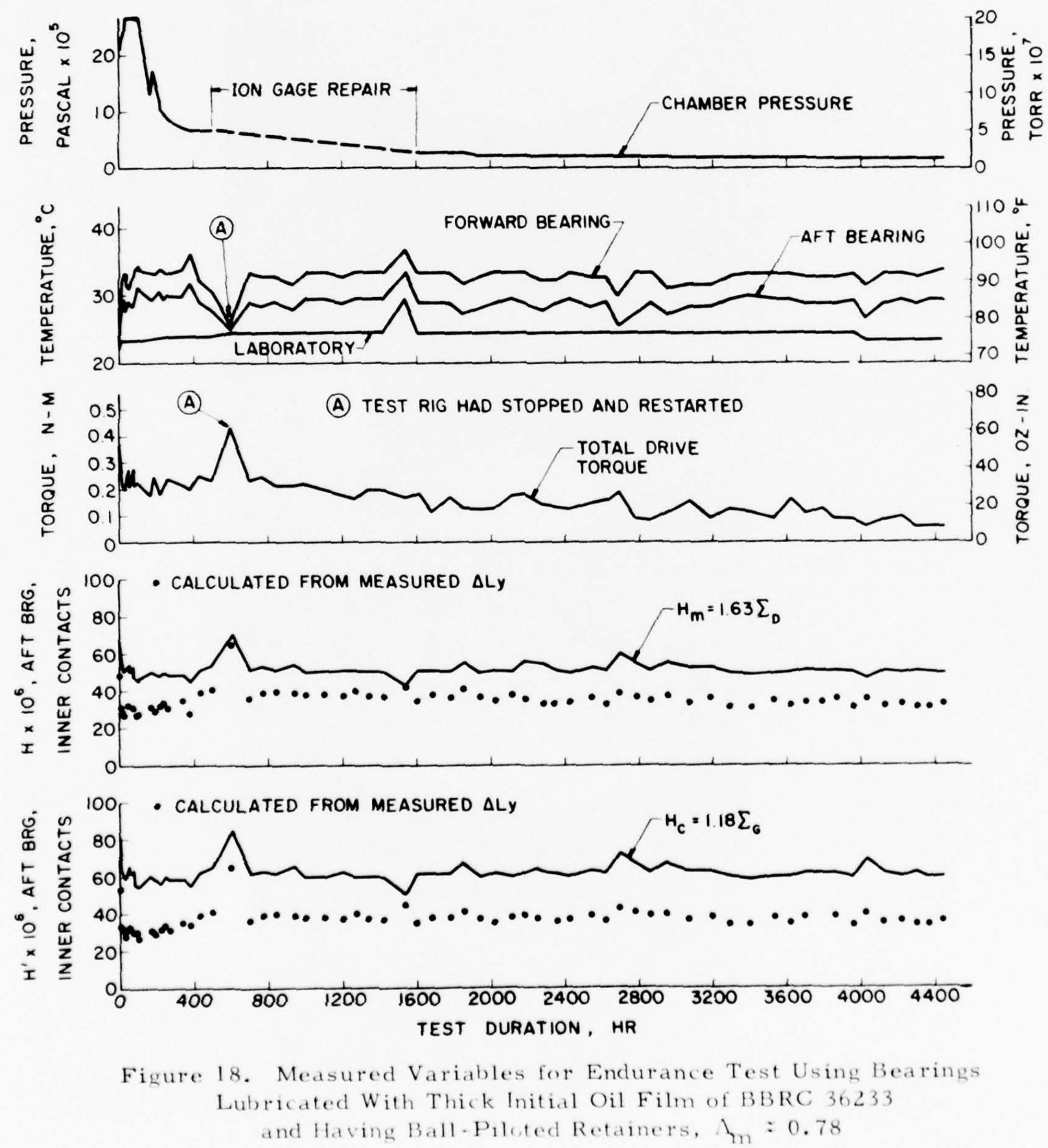

Figure 18. Measured Variables for Endurance Test Using Bearings Lubricated With Thick Initial Oil Film of BBRC 36233 and Having Ball-Piloted Retainers, $\Lambda_m \approx 0.78$

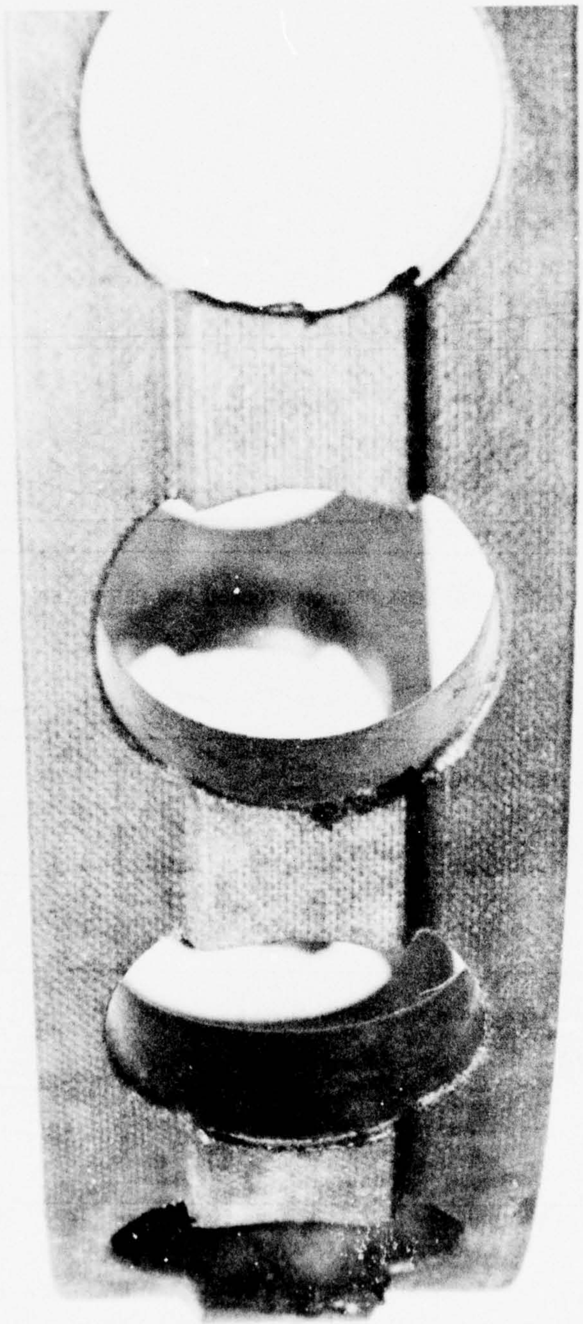

Figure 19. Debris Deposits on Ball-Pocket Tabs of Aft Bearing Retainer

was also debris on the lands of the outer race and in the running track of the outer race. The balls had particles of debris on their surfaces and were somewhat discolored. In other words they were not shiny like a polished ball. There was some debris collected on both sides of the ball track on both the inner and outer races. There was some oil on the lands of the outer race, but all other surfaces of the remaining bearing components appeared to be very dry. The inner race of this bearing had wear marks (similar to fretting) on the surface that mated with the shoulder on the rig shaft. These wear marks are shown in Figure 20 and appeared to be caused by the bearing actually moving slightly on the shaft during operation of the rig.

For the forward bearing used in this test, all components appeared to be very dry of oil. There was some buildup of debris on the ball-pocket tabls of the retainer, but not nearly the amount as seen on the aft bearing discussed above. Buildup appeared to be fairly evenly distributed on all ball-pocket tabs. The balls in this bearing were also discolored and had a dirty appearance. There was no visible pitting, spalling, or flaking in either of these bearings. In fact, the rubbing tracks in the races did not show any wear and after wiping the debris away they were found to be bright and shiny with the original grinding marks still showing with no apparent wear.

The aft bearing from Test No. 2 had a small amount of oil on the lands of the outer race. All other components appeared to be very dry of oil. There was a small amount of debris on both sides of the ball tracks in the inner and outer races, but very little debris buildup on the ball-pocket tabs of the retainer. The balls from this bearing were also discolored with a stained or dirty appearance. The forward bearing used in Test No. 2 had very little oil on any of the components and also displayed very little wear debris. The balls did have the same discolored appearance. In general this bearing was very clean. There was no evidence of pitting, spalling, or flaking in either of these bearings. Again, the grinding marks in the running tracks of the races were plainly visible without any apparent wear.

Both the aft and forward bearings used in Test No. 3 displayed an ample amount of oil on all components after 4,440 hours of testing. Figure 21 shows the inner race of the aft bearing as it appeared when the test was completed. Note the streaks of oil buildup that were left on the race as it was removed from the assembled bearing. These streaks were caused by a large amount of oil being collected at the ball-race conjunctions. As the balls rolled along the race in the axial direction during disassembly, the streaks of

Figure 20. Wear Marks on Inner Race of Aft Bearing

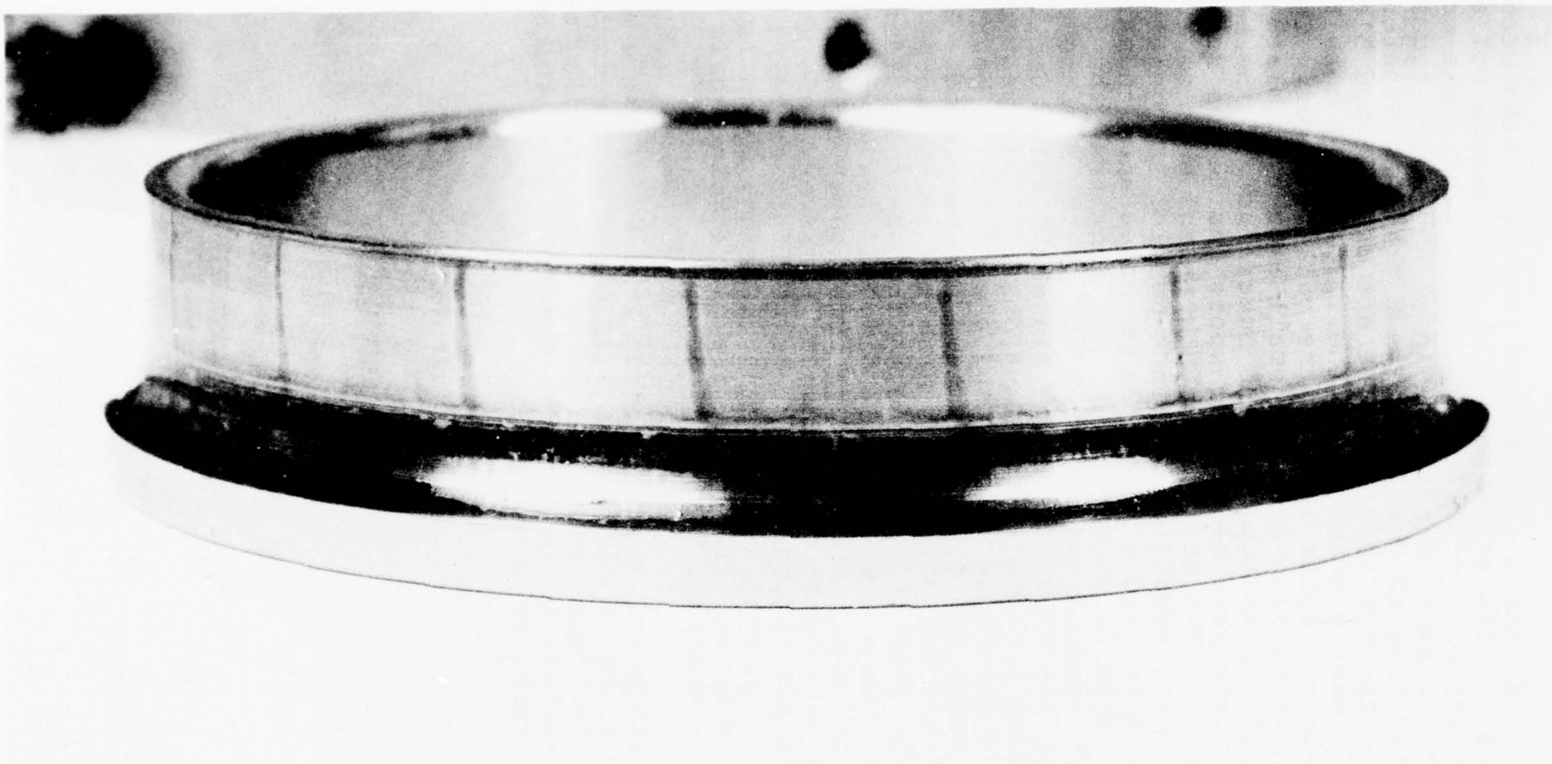

Figure 21. Ball Track in Inner Race of Aft Bearing

oil were left intact on the race surface. Also seen on the inner race, is the large amount of dark debris collected on both sides of the ball track. This debris was collected very much the same on both the inner and outer races of both the aft and forward bearings with a somewhat greater amount visible on the aft bearing. The aft bearing had black stains and debris embedded in the surfaces of both the ball-pocket tabs and outer land of the ball retainer. The forward bearing also showed a small amount of black debris on the ball-pocket tabs and the outer land of the ball retainer. There were black wear spots inside the retainer ball-pockets. The balls in both bearings were covered with oil containing particles of black debris. In general, the forward bearing had a much cleaner appearance than the aft bearing. This is attributed to the fact that the inner ring of the aft bearing had a wear track on the surface that mated with the shoulder on the rig shaft very much the same as discussed previously for Test No. 1. This wear track can be seen in Figure 22 which shows the assembled bearing as it appeared after the 4,440-hour test was completed. It is believed that the wear debris from the slight movement between the inner race and the mating shaft shoulder, for both this test and Test No. 1, did find its way into the bearings and cause the "working components" of the bearings to have a very "dirty" appearance. However, this wear debris was not analyzed to determine if it contained metallic particles. In both tests the aft bearing, which is where the wear occurred, displayed more black wear debris than the forward bearing.

For the bearings employed in Test No. 3 there was no visible pitting, spalling, or wear noted in the running tracks of the races or on the balls. The grinding marks remained visible in the running tracks of both the inner and outer races of both bearings.

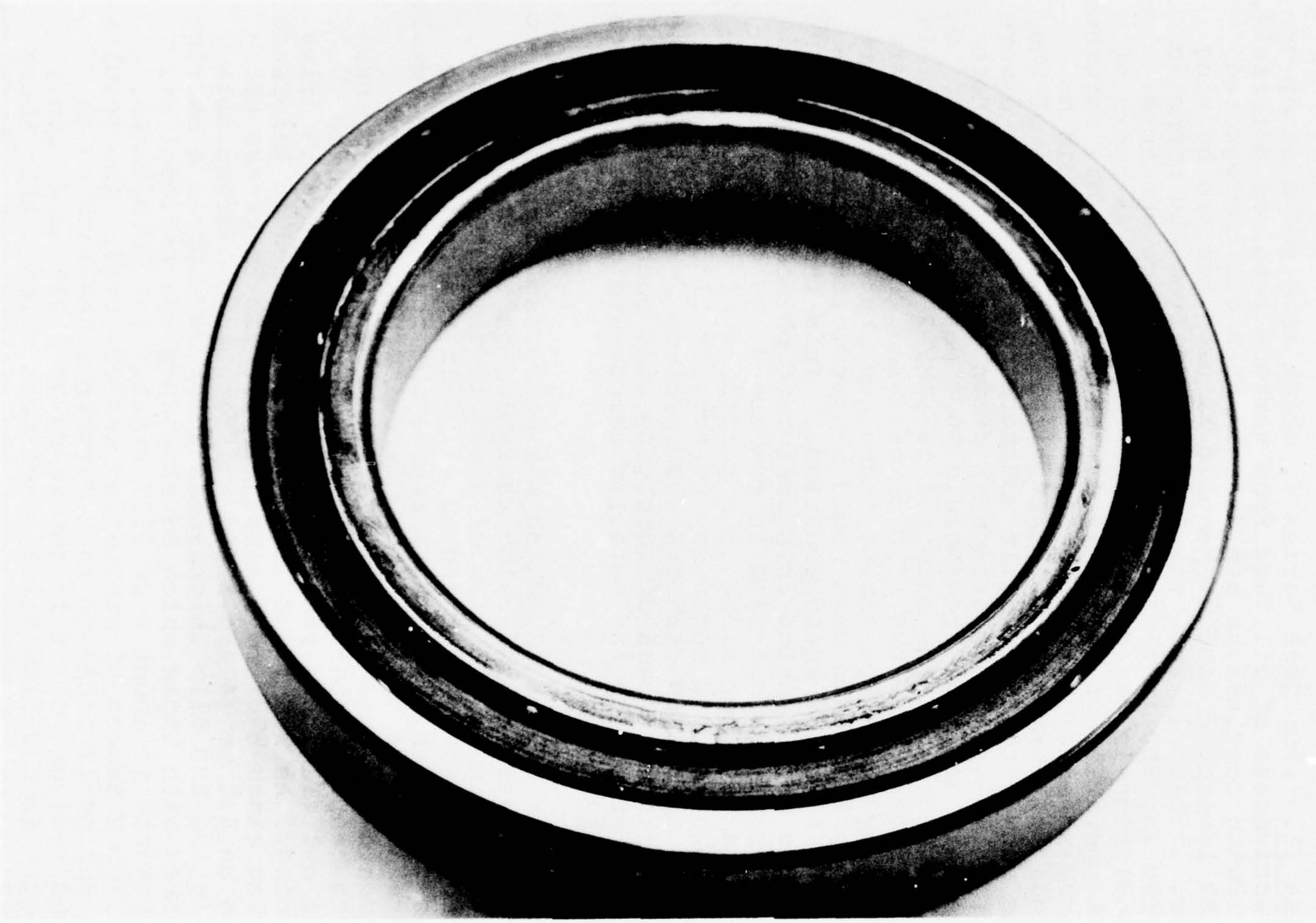

Figure 22. Assembled Aft Bearing Showing Wear on Inner Race

SECTION VI

CONCLUSIONS

Task II. From Test Series I, which involved a study of the effectiveness of oil feed from various retainer materials, the following conclusions are drawn: When oil-impregnated Grade LBB laminated cotton phenolic is used for the bearing retainer, little, if any, oil is released from the phenolic to provide for EHD lubrication of the ball-race contacts. This conclusion is based on the fact that after 24 hours of constant operation of initially dry bearings with oil-saturated retainers, no axial displacement of the bearings could be detected. Furthermore, post-test visual examinations of the bearings showed that the balls and races of the bearings were still dry. The same conclusion is drawn for the case where the retainers are made from Meldin 9000 porous polyimide. On the other hand, when the material is 64HV-2 porous nylon, oil feed does occur. The EHD film thicknesses which develop are measurable by the displacement technique, but in general are considerably less than what would be predicted by the theoretical EHD film thickness equations. This indicates that although the porous nylon is superior to phenolic or porous polyimide as far as oil feed is concerned, an oil-saturated porous nylon retainer alone is not capable of feeding a sufficient amount of oil necessary to provide flooded inlet conditions at the EHD contacts. To provide for fully-flooded inlets, additional means of supplying oil, such as an initial application of oil to the metallic components of the bearings, will be necessary.

From Test Series II, which was concerned with the effect of retainer/bearing processing variables on the ability of bearings to form an oil film of adequate thickness, it is concluded that when the residual oil film remaining on the bearings is too thin, a severe starvation condition may result. For space hardware applications where low bearing torque requirements are necessary, it is common practice to "rinse-back" the bearing after the oil is applied, to thin the residual oil film thus reducing the torque required to drive the bearing. As was found in the study reported in Reference 8, bearing rinse-back to a residual oil film thickness estimated to be 0.1 μm (4 μin.) did not result in a measurable change in the EHD film thicknesses which develop in a bearing as compared with those which develop when the residual oil film thickness is about 4 μm (157 μin.). For both cases, however, some degree of starvation was observed. The results of Test Series II indicate that further rinse-back to an estimated residual oil film thickness of 0.025 μm (1 μin.), will result in EHD film thicknesses much thinner than those which develop in the bearing when the residual oil film thickness is 0.1 μm (4 μin.). Accordingly, the degree of starvation is much greater. In summary, it is concluded that for BBRC 36233, which was the only oil subjected to tests in Test Series II, the residual oil film thickness on the bearings should not be less than about 0.1 μm (4 μin.) if maximum EHD film thicknesses are to be obtained during bearing operation.

Test Series III involved a determination of the effects of frequency and extremes of temperature variation on the formation and maintenance of EHD films in DMA bearings. From the results of the tests conducted in this test series, it is concluded that for the range of temperature excursions and frequencies investigated, there are no irreversible effects on lubricant/bearing performance. That is, as the bearing temperature increases due to an increase in the environmental temperature, the EHD film thickness naturally decreases as a result of the decreased oil viscosity. As the environmental and bearing temperature decrease, the EHD film thickness increases again. During these temperature excursions, the EHD film thickness increases and decreases along a single curve, and the film thickness data are completely correlated by the dimensionless material-load-velocity parameters of Grubin and Dowson. Moreover, during these limited tests performed with BBRC 36233, it appears that over the complete range of temperature excursions, the inlet conditions at the EHD contacts within the bearing are flooded.

In Test Series IV, which involved a study of flooded and starved EHD conditions using the SwRI optical EHD tester, several conclusions may be drawn. The experiments where EHD film thicknesses were measured optically for oil coatings of various thicknesses on glass disks do not support the EHD film thickness measurements made in Task II using the axial displacement technique and actual bearings. With the glass disks, an initial oil film coating with a thickness less than 11.02 μm (434 μin.) resulted in severely starved EHD contact conditions, to the point where the optical interference patterns showed almost no EHD film thickness development. On the other hand, from axial displacement measurements in actual bearings obtained in a previous study[8] and in Task II, an initial oil film coating in the range of 0.1 to 4 μm (4 to 157 μin.) produced EHD films easily measurable and sometimes large enough to indicate fully flooded conditions. The reason for this discrepancy is not known at the present time, but it is believed that the most likely explanation is that the actual residual oil film thickness coatings on the bearings are thicker than those which have been estimated. Other contributing factors may have been the possibility of poorer oil wettability of the glass disks, and centrifugal forces tending to thin the oil coating on the disks.

The experiments with the one-ball retainer models and the SwRI optical EHD tester support, in general, the retainer feed studies conducted with the bearings in Task II. The conclusions are that the Grade LBB phenolic does not result in oil feed to the ball sufficient to develop an EHD film that can be measured optically. The Meldin 9000 porous polyimide material results in a slight amount of oil feed to the ball, enough to be able to detect with the optical interference technique, but not enough to separate the contact surfaces completely. The 64HV-2 porous nylon results in more oil feed than the porous polyimide, although there was not enough oil to create a flooded conjunction. The feed from this material began as soon as the test was initiated and after several minutes of operation side lobes were apparent, but a center strip in the ball-disk conjunction was wiped clean of oil. In conclusion, the phenolic showed no oil feed, the porous polyimide had limited oil feed, and the porous nylon displayed the best oil feed, but not enough for flooded lubrication.

For the experiments with the multi-ball retainer models using the SwRI optical EHD tester, there was not sufficient oil feed from either the Meldin 9000 porous polyimide or Nylasint 64HV-2 porous nylon to cause any separation in the ball-disk contacts. This is not in agreement with the one-ball model experiments and may have resulted because of the different loads and different conformity of the mating surfaces for the two model experiments. Control of the loading and conformity was inherently not as good when using the multi-ball models.

The ball spacing experiments using the Nylasint M-4 porous nylon in the SwRI optical EHD tester showed that for three different ball spacings the ball-disk conjunction film thicknesses were very nearly the same for equivalent material-velocity-load parameters. This leads to the conclusion, at least for the range of conditions covered by these experiments, that the leading ball does not tend to push the oil out of the running track and thus restrict the oil supply to the trailing ball resulting in oil starvation. Of course it should be realized that in these experiments, as would be likely in an actual bearing, the entire complement of balls do not necessarily run along exactly the same track. In fact, there may be as many different tracks as there are balls, thus having many tracks very closely spaced or even partially overlapping.

Task III. For these long-duration tests the low concentration of antiwear additive gave lower torque requirements. On the other hand, the bearings exhibiting the lowest torque requirements (low concentration of antiwear additive) had larger EHD film thicknesses than bearings lubricated with the same base oil containing a higher concentration of antiwear additive. However, the film thicknesses for both oils were well below those predicted by the theoretical equations of Grubin and Dowson for flooded isothermal conditions. As expected, the higher viscosity oil with the higher concentration of antiwear additive exhibited significantly larger film thicknesses and torque requirements, although these film thicknesses remained well below those predicted by Dowson and Grubin. These data in conjunction with the fact that no wear, pitting, or spalling on the running tracks of the balls or races were visible in the post-test inspections, lead to the conclusion that the operation of the bearings for all three tests was in the full EHD regime, although flooded lubrication was not present.

Comparing the torque data for the first 4,440 hours of operation, using BBRC 36233 and ball-piloted retainers with the same oil and outer-land-riding retainers shows that the ball-piloted retainers in general required significantly less torque, especially toward the end of the 4,440-hour period. Since none of the long-duration tests were halted because of bearing seizure, as happened in the earlier program,[8] it is concluded that the new ball-piloted retainer design alleviated the problem. Also it was observed in this program as well as in the previous study, that problems of bearing operation for oil lubricated DMA bearings are more likely to be related to retainer design and/or wear than ball and race wear.

The primary difference between the two long-duration tests using Apiezon A (Test No. 1 and Test No. 2) was the excessive buildup of debris on the ball-pocket tabs of the aft-bearing retainer for the test with the high concentration of antiwear additive (Test No. 1). Both the bearings in the high antiwear additive test exhibited more debris than those employed in the low antiwear additive test, but this can probably be partially attributed to the wear between the inner race of the aft bearing and the mating shoulder of the test rig shaft which occurred during performance of the test.

From the measurements of the EHD film thicknesses in the bearings using the race displacement technique, the Λ_m ratio for the tests where Apiezon A with additives was employed was as low as 0.21. Yet examination of the bearing races and balls indicated that little, if any, rubbing wear occurred. It is thus concluded that lubrication was very nearly in the full EHD regime for these tests. Consequently, it has not been possible to determine the effect of Λ_m on the bearing-lubricant system life. However, it has been established from these tests that for periods of 4,440 hours, satisfactory operation from an EHD film standpoint can be attained at Λ_m values as low as 0.21 even with bearings having a thin initial oil coating and very rough bearing races of 0.406 μm (16 μin.), if ball-piloted retainers are employed.

APPENDIX

TASK III DATA

ENDURANCE TEST NO. 1
OIL = APIEZON A + ANTI + 5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(N) = 890
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROMETERS

TEST TIME (HR)	PRESSURE (PASCAL)	TORQUE (N-M)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS: TEMPA (C)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS: TEMPF (C)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
0	2.1E-04	.092	25.00	22.8	.078	.092	.215	.069	.082	.181	22.8	.078	.092	.215	.069	.082	.181
5	2.3E-04	.085	25.00	24.4	.072	.072	.198	.063	.063	.168	25.6	.068	.068	.187	.060	.060	.159
23	2.4E-04	.092	25.00	28.9	.058	.058	.160	.052	.052	.138	33.3	.048	.048	.131	.044	.044	.115
30	2.7E-04	.092	25.00	28.9	.058	.058	.160	.052	.052	.138	29.4	.057	.057	.156	.051	.051	.135
48	2.7E-04	.141	25.00	27.2	.063	.063	.173	.056	.056	.148	29.4	.057	.057	.156	.051	.051	.135
54	2.7E-04	.134	25.00	27.2	.063	.063	.173	.056	.056	.148	29.4	.057	.057	.156	.051	.051	.135
72	2.7E-04	.141	25.00	27.8	.061	.061	.168	.055	.055	.144	29.4	.057	.057	.156	.051	.051	.135
78	2.7E-04	.141	25.00	28.3	.060	.060	.164	.053	.053	.141	30.0	.055	.055	.152	.050	.050	.132
96	2.7E-04	.134	25.00	29.4	.057	.057	.156	.051	.051	.135	31.1	.053	.053	.145	.048	.048	.126
102	2.7E-04	.092	25.00	29.4	.057	.057	.156	.051	.051	.135	31.1	.053	.053	.145	.048	.048	.126
168	1.3E-04	.078	25.00	27.8	.061	.061	.168	.055	.055	.144	30.0	.055	.055	.152	.050	.050	.132
174	1.7E-04	.071	25.00	28.3	.060	.060	.164	.053	.053	.141	30.6	.054	.054	.148	.049	.049	.129
192	1.2E-04	.099	25.00	28.9	.058	.058	.160	.052	.052	.138	30.6	.054	.054	.148	.049	.049	.129
216	1.1E-04	.113	25.00	29.4	.057	.057	.156	.051	.051	.135	31.1	.053	.053	.145	.048	.048	.126
240	1.0E-04	.120	25.00	28.9	.058	.058	.160	.052	.052	.138	31.1	.053	.053	.145	.048	.048	.126
264	8.7E-05	.099	25.00	28.9	.058	.058	.160	.052	.052	.138	31.1	.053	.053	.145	.048	.048	.126
336	7.3E-05	.134	25.00	27.8	.061	.061	.168	.055	.055	.144	30.0	.055	.055	.152	.050	.050	.132
384	6.7E-05	.078	25.00	28.9	.058	.058	.160	.052	.052	.138	31.1	.053	.053	.145	.048	.048	.126
432	6.7E-05	.092	25.00	28.3	.060	.060	.164	.053	.053	.141	30.6	.054	.054	.148	.049	.049	.129
504	6.7E-05	.071	25.00	27.2	.063	.063	.173	.056	.056	.148	29.4	.057	.057	.156	.051	.051	.135
600	1.2E-04	.113	25.00	27.2	.063	.071	.173	.056	.064	.148	29.4	.057	.064	.156	.051	.059	.135
696	1.2E-04	.092	25.00	28.9	.058	.058	.160	.052	.052	.138	31.1	.053	.053	.145	.048	.048	.126
768	1.2E-04	.099	25.00	27.2	.063	.063	.173	.056	.056	.148	28.9	.058	.058	.160	.052	.052	.138
840	1.2E-04	.092	25.00	26.7	.065	.065	.177	.057	.057	.152	28.3	.060	.060	.164	.053	.053	.141
936	1.2E-04	.134	25.00	26.7	.065	.065	.177	.057	.057	.152	28.3	.060	.060	.164	.053	.053	.141
1008	1.2E-04	.099	25.00	27.2	.063	.063	.173	.056	.056	.148	28.9	.058	.058	.160	.052	.052	.138
1104	1.2E-04	.078	25.00	27.2	.063	.063	.173	.056	.056	.148	28.9	.058	.058	.160	.052	.052	.138
1200	1.2E-04	.092	25.00	26.1	.066	.066	.182	.059	.059	.155	27.8	.061	.061	.168	.055	.055	.144
1272	1.2E-04	.099	25.00	26.1	.066	.075	.182	.059	.067	.155	27.8	.061	.069	.168	.055	.062	.144
1344	1.2E-04	.071	25.00	27.2	.063	.063	.173	.056	.056	.148	28.3	.060	.060	.164	.053	.053	.141
1416	1.2E-04	.078	25.00	26.1	.066	.066	.182	.059	.059	.155	27.8	.061	.061	.168	.055	.055	.144
1536	1.2E-04	.042	25.00	31.7	.051	.068	.141	.047	.063	.123	32.8	.049	.065	.135	.045	.060	.118
1608	2.7E-05	.064	25.00	26.1	.066	.066	.182	.059	.059	.155	28.3	.060	.060	.164	.053	.053	.141
1680	2.7E-05	.099	25.00	26.1	.066	.066	.182	.059	.059	.155	27.8	.061	.061	.168	.055	.055	.144
1776	2.7E-05	.106	25.00	26.1	.066	.080	.182	.059	.073	.155	27.8	.061	.074	.168	.055	.067	.144
1848	2.7E-05	.085	25.00	25.6	.068	.068	.187	.060	.060	.159	27.2	.063	.063	.173	.056	.056	.148
1944	2.0E-05	.099	25.00	27.2	.063	.063	.173	.056	.056	.148	28.3	.060	.060	.164	.053	.053	.141
2016	2.0E-05	.085	25.00	27.2	.063	.071	.173	.055	.064	.148	28.9	.058	.066	.160	.052	.060	.138
2112	2.0E-05	.099	25.00	27.2	.063	.063	.173	.056	.056	.148	27.8	.061	.061	.168	.055	.055	.144

ENDURANCE TEST NO. 1
OIL = APIEZON A + ANTI + 5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(N) = 890
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROMETERS

TEST TIME (HR)	PRESSURE (PASCAL)	TORQUE (N-M)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS: TEMPA (C)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS: TEMPF (C)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
2184	2.0E-05	.078	25.00	27.2	.063	.063	.173	.056	.056	.148	28.3	.060	.060	.164	.053	.053	.141
2280	2.0E-05	.085	25.00	25.6	.068	.068	.187	.060	.060	.159	27.2	.063	.063	.173	.056	.056	.148
2352	2.0E-05	.078	25.00	25.6	.068	.068	.187	.060	.060	.159	27.2	.063	.063	.173	.056	.056	.148
2424	2.0E-05	.071	25.00	26.1	.066	.066	.182	.059	.059	.155	27.8	.061	.061	.168	.055	.055	.144
2544	2.0E-05	.092	25.00	25.6	.068	.068	.187	.060	.060	.159	27.2	.063	.063	.173	.056	.056	.148
2616	2.0E-05	.078	25.00	26.1	.066	.075	.182	.059	.067	.155	27.8	.061	.069	.168	.055	.062	.144
2688	2.0E-05	.078	25.00	24.4	.072	.072	.198	.063	.063	.168	26.1	.066	.066	.182	.059	.059	.155
2784	2.0E-05	.078	25.00	25.0	.070	.070	.192	.062	.062	.163	27.8	.061	.061	.168	.055	.055	.144
2856	2.0E-05	.092	25.00	25.6	.068	.068	.187	.060	.060	.159	27.8	.061	.061	.168	.055	.055	.144
2952	2.0E-05	.078	25.00	25.0	.070	.070	.192	.062	.062	.163	25.6	.068	.068	.187	.060	.060	.159
3072	2.0E-05	.042	25.00	25.6	.068	.068	.187	.060	.060	.159	26.1	.066	.066	.182	.059	.059	.155
3192	2.0E-05	.092	25.00	26.1	.066	.066	.182	.059	.059	.155	27.2	.063	.063	.173	.056	.056	.148
3288	2.0E-05	.071	25.00	25.6	.068	.068	.187	.060	.060	.159	27.2	.063	.063	.173	.056	.056	.148
3408	2.0E-05	.092	25.00	27.8	.061	.061	.168	.055	.055	.144	29.4	.057	.057	.156	.051	.051	.135
3528	2.0E-05	.049	25.00	27.2	.063	.063	.173	.056	.056	.148	29.4	.057	.057	.156	.051	.051	.135
3624	2.0E-05	.049	25.00	26.1	.066	.080	.182	.059	.073	.155	27.8	.061	.074	.168	.055	.067	.144
3696	2.0E-05	.071	25.00	26.1	.066	.066	.182	.059	.059	.155	27.8	.061	.061	.168	.055	.055	.144
3792	2.0E-05	.049	25.00	25.6	.068	.068	.187	.060	.060	.159	27.2	.063	.063	.173	.056	.056	.148
3864	2.0E-05	.085	25.00	26.7	.065	.065	.177	.057	.057	.152	27.8	.061	.061	.168	.055	.055	.144
3960	2.0E-05	.071	25.00	26.7	.065	.065	.177	.057	.057	.152	28.9	.058	.058	.160	.052	.052	.138
4032	1.6E-05	.021	25.00	25.6	.068	.068	.187	.060	.060	.159	27.8	.061	.061	.168	.055	.055	.144
4128	1.6E-05	.092	25.00	27.2	.063	.077	.173	.056	.070	.148	29.4	.057	.069	.156	.051	.064	.135
4224	1.6E-05	.092	25.00	27.2	.063	.077	.173	.056	.070	.148	29.4	.057	.069	.156	.051	.064	.135
4296	1.6E-05	.049	25.00	27.2	.063	.071	.173	.056	.064	.148	28.9	.058	.066	.160	.052	.060	.138

ENDURANCE TEST NO. 1
OIL = APIEZON A + ANTI + 5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(LB) = 200
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROINCHES

TEST TIME (HR)	PRESSURE (TORR)	TORQUE (OZ-IN)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS: TEMPA (F)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS: TEMPF (F)	BHCFI	XCENI	TCENI	BHMFI	XMINI	THINI
0	1.6E-06	13.00	25.00	73.0	3.08	3.61	8.48	2.70	3.22	7.13	73.0	3.08	3.61	8.48	2.70	3.22	7.13
5	1.7E-06	12.00	25.00	76.0	2.83	2.83	7.79	2.50	2.50	6.60	78.0	2.68	2.68	7.37	2.38	2.38	6.27
23	1.8E-06	13.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	92.0	1.88	1.88	5.17	1.72	1.72	4.53
30	2.0E-06	13.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	85.0	2.23	2.23	6.14	2.01	2.01	5.30
48	2.0E-06	20.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	85.0	2.23	2.23	6.14	2.01	2.01	5.30
54	2.0E-06	19.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	85.0	2.23	2.23	6.14	2.01	2.01	5.30
72	2.0E-06	20.00	25.00	82.0	2.41	2.41	6.63	2.15	2.15	5.69	85.0	2.23	2.23	6.14	2.01	2.01	5.30
78	2.0E-06	20.00	25.00	83.0	2.35	2.35	6.46	2.10	2.10	5.55	86.0	2.18	2.18	5.98	1.96	1.96	5.18
96	2.0E-06	19.00	25.00	85.0	2.23	2.23	6.14	2.01	2.01	5.30	88.0	2.07	2.07	5.70	1.87	1.87	4.95
102	2.0E-06	13.00	25.00	85.0	2.23	2.23	6.14	2.01	2.01	5.30	88.0	2.07	2.07	5.70	1.87	1.87	4.95
168	1.0E-06	11.00	25.00	82.0	2.41	2.41	6.63	2.15	2.15	5.69	86.0	2.18	2.18	5.98	1.96	1.96	5.18
174	1.3E-06	10.00	25.00	83.0	2.35	2.35	6.46	2.10	2.10	5.55	87.0	2.12	2.12	5.84	1.92	1.92	5.06
192	9.0E-07	14.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	87.0	2.12	2.12	5.84	1.92	1.92	5.06
216	8.0E-07	16.00	25.00	85.0	2.23	2.23	6.14	2.01	2.01	5.30	88.0	2.07	2.07	5.70	1.87	1.87	4.95
240	7.5E-07	17.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	88.0	2.07	2.07	5.70	1.87	1.87	4.95
264	6.5E-07	14.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	88.0	2.07	2.07	5.70	1.87	1.87	4.95
336	5.5E-07	19.00	25.00	82.0	2.41	2.41	6.63	2.15	2.15	5.69	86.0	2.18	2.18	5.98	1.96	1.96	5.18
384	5.0E-07	11.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	88.0	2.07	2.07	5.70	1.87	1.87	4.95
432	5.0E-07	13.00	25.00	83.0	2.35	2.35	6.46	2.10	2.10	5.55	87.0	2.12	2.12	5.84	1.92	1.92	5.06
504	5.0E-07	10.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	85.0	2.23	2.23	6.14	2.01	2.01	5.30
600	9.0E-07	16.00	25.00	81.0	2.47	2.80	6.80	2.21	2.54	5.83	85.0	2.23	2.53	6.14	2.01	2.31	5.30
696	9.0E-07	13.00	25.00	84.0	2.29	2.29	6.29	2.06	2.06	5.42	88.0	2.07	2.07	5.70	1.87	1.87	4.95
768	9.0E-07	14.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	84.0	2.29	2.29	6.29	2.06	2.06	5.42
840	9.0E-07	13.00	25.00	80.0	2.54	2.54	6.99	2.26	2.26	5.97	83.0	2.35	2.35	6.46	2.10	2.10	5.55
936	9.0E-07	19.00	25.00	80.0	2.54	2.54	6.99	2.26	2.26	5.97	83.0	2.35	2.35	6.46	2.10	2.10	5.55
1008	9.0E-07	14.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	84.0	2.29	2.29	6.29	2.06	2.06	5.42
1104	9.0E-07	11.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	84.0	2.29	2.29	6.29	2.06	2.06	5.42
1200	9.0E-07	13.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	82.0	2.41	2.41	6.63	2.15	2.15	5.69
1272	4.0E-07	14.00	25.00	79.0	2.61	2.94	7.18	2.32	2.64	6.12	82.0	2.41	2.71	6.63	2.15	2.46	5.69
1344	9.0E-07	10.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	83.0	2.35	2.35	6.46	2.10	2.10	5.55
1416	9.0E-07	11.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	82.0	2.41	2.41	6.63	2.15	2.15	5.69
1536	9.0E-07	6.00	25.00	89.0	2.02	2.67	5.56	1.83	2.48	4.84	91.0	1.93	2.54	5.30	1.75	2.37	4.63
1608	2.0E-07	9.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	83.0	2.35	2.35	6.46	2.10	2.10	5.55
1680	2.0E-07	14.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	82.0	2.41	2.41	6.63	2.15	2.15	5.69
1776	2.0E-07	15.00	25.00	79.0	2.61	3.15	7.18	2.32	2.86	6.12	82.0	2.41	2.91	6.63	2.15	2.66	5.69
1848	2.0E-07	12.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	81.0	2.47	2.47	6.80	2.21	2.21	5.83
1944	1.5E-07	14.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	83.0	2.35	2.35	6.46	2.10	2.10	5.55
2016	1.5E-07	12.00	25.00	81.0	2.47	2.80	6.80	2.21	2.53	5.83	84.0	2.29	2.59	6.29	2.06	2.36	5.42
2112	1.5E-07	14.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	82.0	2.41	2.41	6.63	2.15	2.15	5.69

ENDURANCE TEST NO. 1
OIL = APIEZON A + ANTI + 5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(LB) = 200
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROINCHES

TEST TIME (HR)	PRESSURE (TORR)	TORQUE (OZ-IN)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS: TEMPA (F)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS: TEMPF (F)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
2184	1.5E-07	11.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	83.0	2.35	2.35	6.46	2.10	2.10	5.55
2280	1.5E-07	12.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	81.0	2.47	2.47	6.80	2.21	2.21	5.83
2352	1.5E-07	11.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	81.0	2.47	2.47	6.80	2.21	2.21	5.83
2424	1.5E-07	10.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	82.0	2.41	2.41	6.63	2.15	2.15	5.69
2544	1.5E-07	13.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	81.0	2.47	2.47	6.80	2.21	2.21	5.83
2616	1.5E-07	11.00	25.00	79.0	2.61	2.94	7.18	2.32	2.64	6.12	82.0	2.41	2.71	6.63	2.15	2.46	5.69
2688	1.5E-07	11.00	25.00	76.0	2.83	2.83	7.79	2.50	2.50	6.60	79.0	2.61	2.61	7.18	2.32	2.32	6.12
2784	1.5E-07	11.00	25.00	77.0	2.75	2.75	7.58	2.44	2.44	6.43	82.0	2.41	2.41	6.63	2.15	2.15	5.69
2856	1.5E-07	13.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	82.0	2.41	2.41	6.63	2.15	2.15	5.69
2952	1.5E-07	11.00	25.00	77.0	2.75	2.75	7.58	2.44	2.44	6.43	78.0	2.68	2.68	7.37	2.39	2.38	6.27
3072	1.5E-07	6.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	79.0	2.61	2.61	7.18	2.32	2.32	6.12
3192	1.5E-07	13.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	81.0	2.47	2.47	6.80	2.21	2.21	5.83
3288	1.5E-07	10.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	81.0	2.47	2.47	6.80	2.21	2.21	5.83
3408	1.5E-07	13.00	25.00	82.0	2.41	2.41	6.63	2.15	2.15	5.69	85.0	2.23	2.23	6.14	2.01	2.01	5.30
3528	1.5E-07	7.00	25.00	81.0	2.47	2.47	6.80	2.21	2.21	5.83	85.0	2.23	2.23	6.14	2.01	2.01	5.30
3624	1.5E-07	7.00	25.00	79.0	2.61	3.15	7.18	2.32	2.86	6.12	82.0	2.41	2.91	6.63	2.15	2.66	5.69
3696	1.5E-07	10.00	25.00	79.0	2.61	2.61	7.18	2.32	2.32	6.12	82.0	2.41	2.41	6.63	2.15	2.15	5.69
3792	1.5E-07	7.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	81.0	2.47	2.47	6.80	2.21	2.21	5.83
3864	1.5E-07	12.00	25.00	80.0	2.54	2.54	6.99	2.26	2.26	5.97	82.0	2.41	2.41	6.63	2.15	2.15	5.69
3960	1.5E-07	10.00	25.00	80.0	2.54	2.54	6.99	2.26	2.26	5.97	84.0	2.29	2.29	6.29	2.06	2.06	5.42
4032	1.2E-07	3.00	25.00	78.0	2.68	2.68	7.37	2.38	2.38	6.27	82.0	2.41	2.41	6.63	2.15	2.15	5.69
4128	1.2E-07	13.00	25.00	81.0	2.47	3.03	6.80	2.21	2.75	5.83	85.0	2.23	2.73	6.14	2.01	2.50	5.30
4224	1.2E-07	13.00	25.00	81.0	2.47	3.03	6.80	2.21	2.75	5.83	85.0	2.23	2.73	6.14	2.01	2.50	5.30
4296	1.2E-07	7.00	25.00	81.0	2.47	2.80	6.80	2.21	2.53	5.83	84.0	2.29	2.59	6.29	2.06	2.36	5.42

ENDURANCE TEST NO. 2
OIL = APIEZON A + ANTI + 0.5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(N) = 890
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROMETERS

				AFT BEARING, INNER CONTACTS							FORWARD BEARING, INNER CONTACTS						
TEST TIME (HR)	PRESSURE (PASCAL)	TORQUE (N-M)	BASE SPEED (RPM)	TEMPA (C)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	TEMPF (C)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
0	2.1E-04	.028	25.00	22.2	.081	.094	.222	.070	.084	.186	22.2	.081	.094	.222	.070	.084	.186
5	2.3E-04	.064	25.00	31.1	.053	.053	.145	.048	.048	.126	31.7	.051	.051	.141	.047	.047	.123
23	2.4E-04	.028	25.00	28.3	.060	.060	.164	.053	.053	.141	31.1	.053	.053	.145	.048	.048	.126
30	2.7E-04	.035	25.00	27.8	.061	.061	.168	.055	.055	.144	28.9	.058	.058	.160	.052	.052	.138
48	2.7E-04	.056	25.00	27.2	.063	.077	.173	.056	.070	.148	28.3	.060	.073	.164	.053	.066	.141
54	2.7E-04	.120	25.00	27.2	.063	.077	.173	.056	.070	.148	28.9	.058	.071	.160	.052	.065	.138
72	2.7E-04	.056	25.00	27.2	.063	.077	.173	.056	.070	.148	28.9	.058	.071	.160	.052	.065	.138
78	2.7E-04	.035	25.00	27.2	.063	.093	.173	.056	.086	.148	28.9	.058	.086	.160	.052	.080	.138
96	2.7E-04	.042	25.00	28.3	.060	.073	.164	.053	.067	.141	29.4	.057	.070	.156	.051	.064	.135
102	2.7E-04	.049	25.00	27.8	.061	.075	.168	.055	.068	.144	29.4	.057	.069	.155	.051	.064	.135
168	1.3E-04	.035	25.00	27.8	.061	.091	.168	.055	.085	.144	28.9	.058	.087	.160	.052	.081	.138
174	1.7E-04	.056	25.00	27.8	.061	.075	.168	.055	.068	.144	28.9	.059	.071	.160	.052	.065	.138
192	1.2E-04	.035	25.00	27.8	.061	.094	.168	.055	.087	.144	28.9	.059	.089	.160	.052	.083	.138
216	1.1E-04	.042	25.00	28.3	.060	.082	.164	.053	.075	.141	29.4	.057	.077	.156	.051	.072	.135
240	1.0E-04	.028	25.00	27.8	.061	.075	.168	.055	.068	.144	28.9	.058	.071	.160	.052	.065	.138
264	8.7E-05	.056	25.00	27.8	.061	.075	.168	.055	.068	.144	28.9	.059	.071	.160	.052	.065	.138
336	7.3E-05	.035	25.00	27.8	.061	.080	.168	.055	.074	.144	28.9	.058	.076	.160	.052	.070	.138
384	6.7E-05	.028	25.00	27.8	.061	.080	.168	.055	.074	.144	28.9	.058	.076	.160	.052	.070	.138
432	6.7E-05	.035	25.00	27.2	.063	.077	.173	.056	.070	.148	28.9	.058	.071	.160	.052	.065	.138
504	6.7E-05	.028	25.00	28.3	.060	.092	.164	.053	.086	.141	29.4	.057	.088	.156	.051	.082	.135
600	1.2E-04	.035	25.00	26.7	.065	.086	.177	.057	.079	.152	27.8	.061	.082	.168	.055	.075	.144
696	1.2E-04	.042	25.00	26.7	.065	.106	.177	.057	.098	.152	27.8	.061	.100	.168	.055	.094	.144
768	1.2E-04	.049	25.00	26.7	.065	.086	.177	.057	.079	.152	27.8	.061	.082	.168	.055	.075	.144
840	1.2E-04	.056	25.00	26.7	.065	.089	.177	.057	.082	.152	27.8	.061	.085	.168	.055	.078	.144
936	1.2E-04	.085	25.00	26.1	.066	.091	.182	.059	.084	.155	27.8	.061	.084	.168	.055	.078	.144
1008	1.2E-04	.042	25.00	27.2	.063	.085	.173	.056	.078	.148	28.3	.060	.080	.164	.053	.074	.141
1104	1.2E-04	.042	25.00	26.7	.065	.106	.177	.057	.098	.152	27.8	.061	.100	.168	.055	.094	.144
1200	1.2E-04	.042	25.00	26.7	.065	.075	.177	.057	.068	.152	27.2	.063	.073	.173	.056	.067	.148
1272	1.2E-04	.056	25.00	26.7	.065	.089	.177	.057	.082	.152	27.8	.061	.085	.168	.055	.078	.144
1344	1.2E-04	.056	25.00	26.7	.065	.084	.177	.057	.076	.152	27.8	.061	.079	.168	.055	.073	.144
1416	1.2E-04	.042	25.00	26.1	.066	.100	.182	.059	.092	.155	27.8	.061	.092	.168	.055	.085	.144
1536	1.2E-04	.035	25.00	29.4	.057	.076	.156	.051	.070	.135	30.6	.054	.072	.148	.049	.067	.129
1608	2.7E-05	0.000	25.00	25.6	.068	.101	.187	.060	.093	.159	27.2	.063	.094	.173	.056	.087	.148
1680	2.7E-05	.028	25.00	26.1	.066	.088	.182	.059	.081	.155	27.2	.063	.084	.173	.056	.077	.148
1776	2.7E-05	.042	25.00	25.6	.068	.104	.187	.060	.096	.159	26.7	.065	.098	.177	.057	.091	.152
1848	2.7E-05	.035	25.00	25.6	.068	.110	.187	.060	.102	.159	27.2	.063	.101	.173	.056	.094	.148
1944	2.7E-05	.014	25.00	27.2	.063	.092	.173	.056	.085	.148	27.8	.061	.090	.168	.055	.083	.144
2016	2.7E-05	.064	25.00	27.2	.063	.098	.173	.056	.091	.148	27.8	.061	.095	.168	.055	.089	.144
2112	2.7E-05	.064	25.00	26.7	.065	.086	.177	.057	.079	.152	27.2	.063	.084	.173	.056	.077	.148

ENDURANCE TEST NO. 2
OIL = APIEZON A + ANTI + 0.5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(N) = 890
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROMETERS

				AFT BEARING, INNER CONTACTS							FORWARD BEARING, INNER CONTACTS						
TEST TIME (HR)	PRESSURE (PASCAL)	TORQUE (N-M)	BASE SPEED (RPM)	TEMPA (C)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	TEMPF (C)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
2184	2.7E-05	.06	25.00	26.7	.065	.083	.177	.057	.076	.152	27.2	.063	.081	.173	.056	.074	.148
2280	2.7E-05	.042	25.00	25.6	.068	.087	.187	.060	.079	.159	26.7	.065	.083	.177	.057	.076	.152
2352	2.7E-05	.021	25.00	25.6	.068	.101	.187	.060	.093	.159	26.7	.065	.096	.177	.057	.088	.152
2424	2.0E-05	.021	25.00	26.7	.065	.100	.177	.057	.092	.152	27.2	.063	.097	.173	.056	.090	.148
2544	2.0E-05	.028	25.00	25.6	.068	.101	.187	.060	.093	.159	26.7	.065	.096	.177	.057	.088	.152
2616	2.0E-05	.042	25.00	26.1	.066	.091	.182	.059	.083	.155	27.2	.063	.086	.173	.056	.079	.148
2688	2.0E-05	.028	25.00	25.0	.070	.103	.192	.062	.095	.163	26.1	.066	.097	.182	.059	.090	.155
2784	2.0E-05	.049	25.00	25.0	.070	.102	.192	.062	.094	.163	25.6	.068	.100	.187	.060	.092	.159
2856	2.0E-05	.064	25.00	26.7	.065	.087	.177	.057	.079	.152	28.3	.060	.080	.164	.053	.074	.141
2952	2.0E-05	.028	25.00	23.3	.076	.106	.209	.067	.096	.176	23.3	.076	.106	.209	.067	.096	.176
3072	2.0E-05	.049	25.00	25.0	.070	.083	.192	.062	.075	.163	25.0	.070	.083	.192	.062	.075	.163
3192	2.0E-05	.064	25.00	27.2	.063	.104	.173	.056	.097	.148	28.3	.060	.099	.164	.053	.092	.141
3288	2.0E-05	.042	25.00	26.1	.066	.088	.182	.059	.081	.155	27.2	.063	.084	.173	.056	.077	.148
3408	2.0E-05	.099	25.00	28.9	.058	.080	.160	.052	.074	.138	30.0	.055	.076	.152	.050	.070	.132
3528	2.0E-05	.035	25.00	28.9	.058	.077	.160	.052	.071	.138	29.4	.057	.075	.156	.051	.069	.135
3624	2.0E-05	.028	25.00	25.6	.068	.090	.187	.060	.082	.159	27.2	.063	.083	.173	.056	.077	.148
3696	2.0E-05	.064	25.00	25.6	.068	.093	.187	.060	.085	.159	27.2	.063	.086	.173	.056	.079	.148
3792	2.0E-05	.042	25.00	27.2	.063	.093	.173	.056	.086	.148	28.3	.060	.088	.164	.053	.082	.141
3864	2.0E-05	.042	25.00	26.7	.065	.083	.177	.057	.076	.152	27.2	.063	.081	.173	.056	.074	.148
3960	2.0E-05	.028	25.00	27.2	.063	.084	.173	.056	.077	.148	27.8	.061	.082	.168	.055	.076	.144
4032	1.6E-05	.085	25.00	25.0	.070	.092	.192	.062	.084	.163	26.7	.065	.085	.177	.057	.078	.152
4128	1.6E-05	.078	25.00	27.8	.061	.080	.168	.055	.074	.144	28.3	.060	.078	.164	.053	.072	.141
4224	1.6E-05	.028	25.00	27.2	.063	.074	.173	.056	.067	.148	28.9	.058	.068	.160	.052	.062	.138
4296	1.6E-05	.064	25.00	27.2	.063	.085	.173	.056	.078	.148	28.3	.060	.080	.164	.053	.074	.141

ENDURANCE TEST NO. 2
OIL = APIEZON A + ANTI + 0.5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(LB) = 200
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROINCHES

TEST TIME (HR)	PRESSURE (TORR)	TORQUE (OZ-IN)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS TEMPA (F)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS TEMPF (F)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
0	1.6E-06	4.00	25.00	72.0	3.17	3.70	8.73	2.77	3.29	7.32	72.0	3.17	3.70	8.73	2.77	3.29	7.32
5	1.7E-06	9.00	25.00	88.0	2.07	2.07	5.70	1.87	1.87	4.95	89.0	2.02	2.02	5.56	1.83	1.83	4.84
23	1.8E-06	4.00	25.00	83.0	2.35	2.35	6.46	2.10	2.10	5.55	88.0	2.07	2.07	5.70	1.87	1.87	4.95
30	2.0E-06	5.00	25.00	82.0	2.41	2.41	6.63	2.15	2.15	5.69	84.0	2.29	2.29	6.29	2.06	2.06	5.42
48	2.0E-06	8.00	25.00	81.0	2.47	3.01	6.80	2.21	2.74	5.83	83.0	2.35	2.86	6.46	2.10	2.61	5.55
54	2.0E-06	17.00	25.00	81.0	2.47	3.02	6.80	2.21	2.75	5.83	84.0	2.29	2.79	6.29	2.06	2.56	5.42
72	2.0E-06	8.00	25.00	81.0	2.47	3.02	6.80	2.21	2.75	5.83	84.0	2.29	2.73	6.29	2.06	2.56	5.42
78	2.0E-06	5.00	25.00	81.0	2.47	3.68	6.80	2.21	3.39	5.83	84.0	2.29	3.40	6.29	2.06	3.16	5.42
96	2.0E-06	6.00	25.00	83.0	2.35	2.89	6.46	2.10	2.64	5.55	85.0	2.23	2.74	6.14	2.01	2.52	5.30
102	2.0E-06	7.00	25.00	82.0	2.41	2.95	6.63	2.15	2.69	5.69	85.0	2.23	2.73	6.14	2.01	2.51	5.30
168	1.0E-06	5.00	25.00	82.0	2.41	3.60	6.63	2.15	3.33	5.69	84.0	2.29	3.41	6.29	2.06	3.18	5.42
174	1.3E-06	8.00	25.00	82.0	2.41	2.95	6.63	2.15	2.69	5.69	84.0	2.29	2.80	6.29	2.06	2.57	5.42
192	9.0E-07	5.00	25.00	82.0	2.41	3.70	6.63	2.15	3.44	5.69	84.0	2.29	3.52	6.29	2.06	3.28	5.42
216	9.0E-07	6.00	25.00	83.0	2.35	3.21	6.46	2.10	2.96	5.55	85.0	2.23	3.05	6.14	2.01	2.82	5.30
240	7.5E-07	4.00	25.00	82.0	2.41	2.95	6.63	2.15	2.69	5.69	84.0	2.29	2.80	6.29	2.06	2.57	5.42
264	6.5E-07	8.00	25.00	82.0	2.41	2.95	6.63	2.15	2.69	5.69	84.0	2.29	2.80	6.29	2.06	2.57	5.42
336	5.5E-07	5.00	25.00	82.0	2.41	3.16	6.63	2.15	2.90	5.69	84.0	2.29	3.00	6.29	2.06	2.77	5.42
384	5.0E-07	4.00	25.00	82.0	2.41	3.16	6.63	2.15	2.90	5.69	84.0	2.29	3.00	6.29	2.06	2.77	5.42
432	5.0E-07	5.00	25.00	81.0	2.47	3.02	6.80	2.21	2.75	5.83	84.0	2.29	2.79	6.29	2.06	2.56	5.42
504	5.0E-07	4.00	25.00	83.0	2.35	3.64	6.46	2.10	3.39	5.55	85.0	2.23	3.46	6.14	2.01	3.23	5.30
600	9.0E-07	5.00	25.00	80.0	2.54	3.40	6.99	2.26	3.12	5.97	82.0	2.41	3.23	6.63	2.15	2.97	5.69
696	9.0E-07	6.00	25.00	80.0	2.54	4.16	6.99	2.26	3.87	5.97	82.0	2.41	3.94	6.63	2.15	3.69	5.69
768	9.0E-07	7.00	25.00	80.0	2.54	3.40	6.99	2.26	3.12	5.97	82.0	2.41	3.23	6.63	2.15	2.97	5.69
840	9.0E-07	8.00	25.00	80.0	2.54	3.51	6.99	2.26	3.22	5.97	82.0	2.41	3.33	6.63	2.15	3.07	5.69
936	9.0E-07	12.00	25.00	79.0	2.61	3.59	7.18	2.32	3.29	6.12	82.0	2.41	3.32	6.63	2.15	3.06	5.69
1008	9.0E-07	6.00	25.00	81.0	2.47	3.33	6.80	2.21	3.06	5.83	83.0	2.35	3.17	6.46	2.10	2.92	5.55
1104	9.0E-07	6.00	25.00	80.0	2.54	4.16	6.99	2.26	3.87	5.97	82.0	2.41	3.94	6.63	2.15	3.69	5.69
1200	9.0E-07	6.00	25.00	80.0	2.54	2.97	6.99	2.26	2.68	5.97	81.0	2.47	2.89	6.80	2.21	2.62	5.83
1272	9.0E-07	8.00	25.00	80.0	2.54	3.51	6.99	2.26	3.22	5.97	82.0	2.41	3.33	6.63	2.15	3.07	5.69
1344	9.0E-07	8.00	25.00	80.0	2.54	3.29	6.99	2.26	3.01	5.97	82.0	2.41	3.12	6.63	2.15	2.87	5.69
1416	9.0E-07	6.00	25.00	79.0	2.61	3.92	7.18	2.32	3.61	6.12	82.0	2.41	3.62	6.63	2.15	3.36	5.69
1536	9.0E-07	5.00	25.00	85.0	2.23	2.99	6.14	2.01	2.75	5.30	87.0	2.12	2.84	5.84	1.92	2.63	5.06
1608	2.0E-07	0.00	25.00	78.0	2.68	3.99	7.37	2.38	3.67	6.27	81.0	2.47	3.69	6.80	2.21	3.41	5.83
1680	2.0E-07	4.00	25.00	79.0	2.61	3.47	7.18	2.32	3.17	6.12	81.0	2.47	3.29	6.80	2.21	3.02	5.83
1776	2.0E-07	6.00	25.00	78.0	2.68	4.08	7.37	2.38	3.77	6.27	80.0	2.54	3.87	6.99	2.26	3.59	5.97
1848	2.0E-07	5.00	25.00	78.0	2.68	4.32	7.37	2.38	4.00	6.27	81.0	2.47	3.98	6.80	2.21	3.72	5.83
1944	2.0E-07	2.00	25.00	81.0	2.47	3.64	6.80	2.21	3.36	5.83	82.0	2.41	3.55	6.63	2.15	3.28	5.69
2016	2.0E-07	9.00	25.00	81.0	2.47	3.85	6.80	2.21	3.58	5.83	82.0	2.41	3.76	6.63	2.15	3.49	5.69
2112	2.0E-07	9.00	25.00	80.0	2.54	3.39	6.99	2.26	3.10	5.97	81.0	2.47	3.30	6.80	2.21	3.03	5.83

ENDURANCE TEST NO. 2
OIL = APIEZON A + ANTI + 0.5% LEAD NAPH
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THIN
LOAD(LB) = 200
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROINCHES

				AFT BEARING, INNER CONTACTS							FORWARD BEARING, INNER CONTACTS						
TEST TIME (HR)	PRESSURE (TORR)	TORQUE (OZ-IN)	BASE SPEED (RPM)	TEMPA (F)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	TEMPF (F)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
2184	2.0E-07	9.00	25.00	80.0	2.54	3.28	6.99	2.26	3.00	5.97	81.0	2.47	3.20	6.80	2.21	2.93	5.83
2280	2.0E-07	6.00	25.00	78.0	2.68	3.43	7.37	2.38	3.13	6.27	80.0	2.54	3.25	6.99	2.26	2.98	5.97
2352	2.0E-07	3.00	25.00	78.0	2.68	3.97	7.37	2.38	3.66	6.27	80.0	2.54	3.77	6.99	2.26	3.48	5.97
2424	1.5E-07	3.00	25.00	80.0	2.54	3.92	6.99	2.26	3.63	5.97	81.0	2.47	3.82	6.80	2.21	3.55	5.83
2544	1.5E-07	4.00	25.00	78.0	2.68	3.97	7.37	2.38	3.66	6.27	80.0	2.54	3.77	6.99	2.26	3.48	5.97
2616	1.5E-07	6.00	25.00	79.0	2.61	3.58	7.18	2.32	3.28	6.12	81.0	2.47	3.39	6.80	2.21	3.12	5.83
2688	1.5E-07	4.00	25.00	77.0	2.75	4.05	7.58	2.44	3.72	6.43	79.0	2.61	3.83	7.18	2.32	3.54	6.12
2784	1.5E-07	7.00	25.00	77.0	2.75	4.03	7.58	2.44	3.70	6.43	78.0	2.68	3.92	7.37	2.38	3.61	6.27
2856	1.5E-07	9.00	25.00	80.0	2.54	3.42	6.99	2.26	3.13	5.97	83.0	2.35	3.16	6.46	2.10	2.41	5.55
2952	1.5E-07	4.00	25.00	74.0	3.00	4.15	8.24	2.63	3.78	6.94	74.0	3.00	4.15	8.24	2.63	3.78	6.94
3072	1.5E-07	7.00	25.00	77.0	2.75	3.28	7.58	2.44	2.96	6.43	77.0	2.75	3.28	7.59	2.44	2.96	6.43
3192	1.5E-07	9.00	25.00	81.0	2.47	4.09	6.80	2.21	3.81	5.83	83.0	2.35	3.88	6.46	2.10	3.64	5.55
3288	1.5E-07	6.00	25.00	79.0	2.61	3.47	7.18	2.32	3.17	6.12	81.0	2.47	3.29	6.80	2.21	3.02	5.83
3408	1.5E-07	14.00	25.00	84.0	2.29	3.15	6.29	2.06	2.91	5.42	86.0	2.18	2.99	5.98	1.96	2.78	5.18
3528	1.5E-07	5.00	25.00	84.0	2.29	3.03	6.29	2.06	2.80	5.42	85.0	2.23	2.96	6.14	2.01	2.73	5.30
3624	1.5E-07	4.00	25.00	78.0	2.68	3.56	7.37	2.38	3.24	6.27	81.0	2.47	3.28	6.80	2.21	3.01	5.83
3696	1.5E-07	9.00	25.00	78.0	2.68	3.66	7.37	2.38	3.35	6.27	81.0	2.47	3.38	6.80	2.21	3.11	5.83
3792	1.5E-07	6.00	25.00	81.0	2.47	3.66	6.80	2.21	3.38	5.83	83.0	2.35	3.47	6.46	2.10	3.23	5.55
3864	1.5E-07	6.00	25.00	80.0	2.54	3.28	6.99	2.26	3.00	5.97	81.0	2.47	3.20	6.80	2.21	2.93	5.83
3460	1.5E-07	4.00	25.00	81.0	2.47	3.33	6.80	2.21	3.05	5.83	82.0	2.41	3.24	6.63	2.15	2.98	5.69
4032	1.2E-07	12.00	25.00	77.0	2.75	3.63	7.58	2.44	3.30	6.43	80.0	2.54	3.35	6.99	2.26	3.07	5.97
4128	1.2E-07	11.00	25.00	82.0	2.41	3.15	6.63	2.15	2.89	5.69	83.0	2.35	3.07	6.46	2.10	2.83	5.55
4224	1.2E-07	4.00	25.00	81.0	2.47	2.91	6.80	2.21	2.64	5.83	84.0	2.29	2.69	6.29	2.06	2.46	5.42
4296	1.2E-07	9.00	25.00	81.0	2.47	3.33	6.80	2.21	3.06	5.83	83.0	2.35	3.17	6.46	2.10	2.92	5.55

ENDURANCE TEST NO. 3
OIL = 98RC 36233
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THICK
LOAD(N) = 890
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROMETERS

TEST TIME (HR)	PRESSURE (PASCAL)	TORQUE (N-M)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS TEMPA (C)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS TEMPF (C)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
0	2.1E+02	.3 7	25.00	23.3	.206	.371	.567	.176	.340	.465	23.3	.206	.371	.567	.176	.340	.465
5	2.3E+02	.304	25.00	27.8	.166	.240	.458	.143	.216	.378	30.6	.146	.211	.403	.127	.192	.335
23	2.4E+02	.212	25.00	29.4	.154	.221	.424	.133	.199	.351	33.3	.130	.186	.357	.113	.168	.297
30	2.7E+02	.198	25.00	30.0	.150	.188	.413	.130	.167	.343	33.3	.130	.162	.357	.113	.145	.297
48	2.7E+02	.268	25.00	27.8	.166	.224	.458	.143	.200	.378	31.1	.143	.192	.393	.124	.173	.327
56	2.7E+02	.212	25.00	28.9	.158	.220	.435	.136	.198	.360	31.7	.139	.194	.383	.121	.176	.319
72	2.7E+02	.275	25.00	28.3	.162	.208	.446	.140	.185	.369	31.7	.139	.179	.383	.121	.160	.319
78	2.7E+02	.205	25.00	31.1	.143	.208	.393	.124	.188	.327	33.9	.127	.184	.348	.110	.167	.291
96	2.7E+02	.205	25.00	31.7	.139	.187	.383	.121	.169	.319	34.4	.124	.166	.340	.108	.150	.284
102	2.7E+02	.226	25.00	31.1	.143	.188	.393	.124	.169	.327	33.9	.127	.167	.348	.110	.150	.291
168	1.3E+02	.177	25.00	29.4	.154	.218	.424	.133	.196	.351	33.3	.130	.183	.357	.113	.166	.297
174	1.7E+02	.205	25.00	24.4	.154	.212	.424	.133	.190	.351	33.3	.130	.178	.357	.113	.161	.297
192	1.2E+03	.247	25.00	30.0	.150	.205	.413	.130	.184	.343	33.3	.130	.177	.357	.113	.159	.247
216	1.1E+03	.184	25.00	30.6	.146	.221	.403	.127	.200	.335	33.9	.127	.191	.348	.110	.174	.291
240	1.0E+03	.212	25.00	29.4	.154	.235	.424	.133	.213	.351	33.3	.130	.198	.357	.113	.181	.297
264	8.7E+02	.240	25.00	30.0	.150	.216	.413	.130	.195	.343	33.3	.130	.186	.357	.113	.169	.297
336	7.3E+02	.219	25.00	30.0	.150	.249	.413	.130	.227	.343	33.9	.127	.210	.348	.110	.193	.291
384	6.7E+02	.198	25.00	31.7	.139	.186	.383	.121	.167	.319	36.1	.115	.154	.317	.101	.139	.266
432	6.7E+02	.254	25.00	28.9	.158	.273	.435	.136	.250	.360	32.2	.136	.235	.374	.118	.216	.312
504	6.7E+02	.233	25.00	27.8	.166	.289	.458	.143	.264	.378	30.6	.146	.254	.403	.127	.233	.335
600	1.2E+03	.431	25.00	22.2	.218	.455	.600	.186	.419	.491	25.6	.185	.386	.509	.159	.357	.419
696	1.2E-04	.233	25.00	28.9	.158	.255	.435	.136	.232	.360	33.3	.130	.209	.357	.113	.192	.297
768	1.2E-04	.247	25.00	28.3	.162	.277	.446	.140	.253	.369	32.8	.133	.227	.365	.115	.209	.304
840	1.2E-04	.212	25.00	28.9	.158	.280	.435	.136	.257	.360	32.8	.133	.235	.365	.115	.217	.304
936	1.2E-04	.212	25.00	27.8	.166	.274	.458	.143	.250	.378	31.7	.139	.230	.383	.121	.211	.319
1008	1.2E-04	.219	25.00	29.4	.154	.270	.424	.133	.248	.351	33.3	.130	.227	.357	.113	.210	.297
1104	1.2E-04	.198	25.00	29.4	.154	.270	.424	.133	.248	.351	33.3	.130	.227	.357	.113	.210	.297
1200	1.2E-04	.155	25.00	28.9	.158	.256	.435	.136	.234	.360	32.8	.133	.216	.365	.115	.198	.304
1272	1.2E-04	.162	25.00	29.4	.154	.279	.424	.133	.257	.351	33.3	.130	.234	.357	.113	.217	.247
1344	1.2E-04	.198	25.00	29.4	.154	.262	.424	.133	.239	.351	33.3	.130	.220	.357	.113	.202	.297
1416	1.2E-04	.198	25.00	29.4	.154	.262	.424	.133	.239	.351	33.3	.130	.220	.357	.113	.202	.297
1536	1.2E-04	.169	25.00	33.3	.130	.316	.357	.113	.297	.297	36.7	.113	.274	.310	.098	.259	.260
1608	2.7E-05	.184	25.00	28.9	.158	.249	.435	.136	.226	.360	33.3	.130	.204	.357	.113	.187	.297
1680	2.7E-05	.113	25.00	28.9	.158	.264	.435	.136	.241	.360	33.3	.130	.216	.357	.113	.199	.297
1776	2.7E-05	.169	25.00	28.9	.158	.258	.435	.136	.235	.360	33.3	.130	.211	.357	.113	.194	.247
1848	2.7E-05	.134	25.00	27.2	.171	.288	.470	.147	.263	.388	31.7	.139	.235	.383	.121	.217	.319
1944	2.7E-05	.127	25.00	29.4	.154	.260	.424	.133	.238	.351	32.8	.133	.224	.365	.115	.206	.304
2016	2.7E-05	.134	25.00	28.9	.158	.249	.435	.136	.226	.360	33.3	.130	.204	.357	.113	.187	.297
2112	2.7E-05	.184	25.00	29.4	.154	.267	.424	.133	.245	.351	33.3	.130	.224	.357	.113	.207	.297

ENDURANCE TEST NO. 3
OIL = [illegible]
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THICK
LOAD(N) = 890
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROMETERS

				AFT BEARING, INNER CONTACTS							FORWARD BEARING, INNER CONTACTS						
TEST TIME (HR)	PRESSURE (PASCAL)	TORQUE (N-M)	BASE SPEED (RPM)	TEMPA (C)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	TEMPF (C)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
2184	2.7E-05	.184	25.00	28.9	.158	.276	.435	.136	.252	.360	33.3	.130	.226	.357	.113	.208	.297
2280	2.7E-05	.141	25.00	28.3	.162	.258	.446	.140	.234	.369	32.2	.136	.217	.374	.118	.198	.312
2352	2.7E-05	.134	25.00	28.9	.158	.255	.435	.136	.233	.360	32.2	.136	.220	.374	.118	.202	.312
2424	2.0E-05	.127	25.00	29.4	.154	.262	.424	.133	.239	.351	33.3	.130	.220	.357	.113	.202	.297
2544	2.0E-05	.148	25.00	28.3	.162	.277	.446	.140	.253	.369	32.8	.133	.227	.365	.115	.209	.304
2616	2.0E-05	.162	25.00	28.9	.158	.253	.435	.136	.231	.360	32.8	.133	.213	.365	.115	.195	.304
2688	2.0E-05	.191	25.00	25.6	.185	.302	.509	.159	.275	.419	30.0	.150	.246	.413	.130	.225	.343
2784	2.0E-05	.092	25.00	27.2	.171	.289	.470	.147	.263	.388	33.3	.130	.220	.357	.113	.202	.297
2856	2.0E-05	.085	25.00	28.9	.158	.273	.435	.136	.249	.360	33.3	.130	.224	.357	.113	.206	.297
2952	2.0E-05	.113	25.00	27.2	.171	.284	.470	.147	.259	.388	31.1	.143	.238	.393	.124	.218	.327
3072	2.0E-05	.148	25.00	28.3	.162	.260	.446	.140	.236	.369	31.7	.139	.223	.383	.121	.204	.319
3192	2.0E-05	.092	25.00	28.3	.162	.274	.446	.140	.250	.369	31.7	.139	.235	.383	.121	.217	.319
3288	2.0E-05	.127	25.00	29.4	.154	.237	.424	.133	.215	.351	32.8	.133	.204	.365	.115	.167	.304
3408	2.0E-05	.113	25.00	30.0	.150	.239	.413	.130	.218	.343	33.3	.130	.206	.357	.113	.189	.297
3528	2.0E-05	.092	25.00	29.4	.154	.267	.424	.133	.245	.351	33.3	.130	.224	.357	.113	.207	.297
3624	2.0E-05	.162	25.00	29.4	.154	.244	.424	.133	.222	.351	33.3	.130	.205	.357	.113	.188	.297
3696	2.0E-05	.113	25.00	28.9	.158	.266	.435	.136	.242	.360	32.8	.133	.223	.365	.115	.205	.304
3792	2.0E-05	.127	25.00	28.9	.158	.266	.435	.136	.242	.360	32.8	.133	.223	.365	.115	.205	.304
3864	2.0E-05	.092	25.00	28.9	.158	.274	.435	.136	.251	.360	32.8	.133	.230	.365	.115	.212	.304
3960	2.0E-05	.085	25.00	29.4	.154	.238	.424	.133	.216	.351	33.3	.130	.200	.357	.113	.183	.297
4032	1.6E-05	.064	25.00	26.7	.175	.282	.482	.151	.256	.396	31.7	.134	.224	.383	.121	.206	.319
4128	1.6E-05	.085	25.00	28.9	.158	.249	.435	.136	.226	.360	33.3	.130	.204	.357	.113	.187	.297
4224	1.6E-05	.099	25.00	29.4	.154	.256	.424	.133	.234	.351	33.3	.130	.215	.357	.113	.198	.297
4296	1.6E-05	.056	25.00	28.9	.158	.242	.435	.136	.219	.360	32.8	.133	.203	.365	.115	.186	.304

ENDURANCE TEST NO. 3
OIL = 88RC 36233
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THICK
LOAD(LB) = 200
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROINCHES

TEST TIME (HR)	PRESSURE (TORR)	TORQUE (OZ-IN)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS: TEMPA (F)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS: TEMPF (F)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
0	1.6E+00	52.00	25.00	74.0	8.12	14.62	22.34	6.94	13.40	18.30	74.0	8.12	14.62	22.34	6.94	13.40	18.30
5	1.7E+00	43.00	25.00	82.0	6.55	9.45	18.02	5.64	8.52	14.89	87.0	5.77	8.32	15.86	4.99	7.54	13.17
23	1.8E+00	30.00	25.00	85.0	6.06	8.69	16.68	5.24	7.83	13.82	92.0	5.10	7.31	14.04	4.44	6.63	11.71
30	2.0E+00	28.00	25.00	86.0	5.91	7.38	16.27	5.11	6.56	13.49	92.0	5.10	6.37	14.04	4.44	5.70	11.71
48	2.0E+00	38.00	25.00	82.0	6.55	8.80	18.02	5.64	7.88	14.89	88.0	5.62	7.56	15.47	4.87	6.80	12.86
54	2.0E+00	30.00	25.00	84.0	6.22	8.67	17.11	5.37	7.80	14.17	89.0	5.49	7.65	15.10	4.76	6.91	12.56
72	2.0E+00	39.00	25.00	83.0	6.38	8.19	17.56	5.50	7.29	14.52	89.0	5.49	7.05	15.10	4.76	6.31	12.56
78	2.0E+00	29.00	25.00	88.0	5.62	8.19	15.47	4.87	7.40	12.86	93.0	4.98	7.26	13.71	4.34	6.59	11.44
96	2.0E+00	29.00	25.00	89.0	5.49	7.38	15.10	4.76	6.64	12.56	94.0	4.87	6.54	13.39	4.24	5.91	11.19
102	2.0E+00	32.00	25.00	88.0	5.62	7.41	15.47	4.87	6.64	12.86	93.0	4.98	6.56	13.71	4.34	5.91	11.44
168	1.0E+00	25.00	25.00	85.0	6.06	8.57	16.68	5.24	7.73	13.82	92.0	5.10	7.22	14.04	4.44	6.55	11.71
174	1.3E+00	29.00	25.00	85.0	6.06	8.34	16.68	5.24	7.49	13.82	92.0	5.10	7.02	14.04	4.44	6.35	11.71
192	9.0E+00	35.00	25.00	86.0	5.91	8.05	16.27	5.11	7.23	13.49	92.0	5.10	6.95	14.04	4.44	6.28	11.71
216	8.0E+00	26.00	25.00	87.0	5.77	8.71	15.86	4.99	7.89	13.17	93.0	4.98	7.52	13.71	4.34	6.86	11.44
240	7.5E+00	30.00	25.00	85.0	6.06	9.26	16.68	5.24	8.39	13.82	92.0	5.10	7.79	14.04	4.44	7.11	11.71
264	6.5E+00	34.00	25.00	86.0	5.91	8.50	16.27	5.11	7.68	13.49	92.0	5.10	7.34	14.04	4.44	6.67	11.71
336	5.5E+00	31.00	25.00	86.0	5.91	9.79	16.27	5.11	8.95	13.49	93.0	4.98	8.25	13.71	4.34	7.59	11.44
384	5.0E+00	28.00	25.00	89.0	5.49	7.33	15.10	4.76	6.58	12.56	97.0	4.54	6.06	12.48	3.96	5.48	10.46
432	5.0E+00	36.00	25.00	84.0	6.22	10.73	17.11	5.37	9.84	14.17	90.0	5.36	9.24	14.73	4.65	8.52	12.27
504	5.0E+00	33.00	25.00	82.0	6.55	11.37	18.02	5.64	10.39	14.89	87.0	5.77	10.01	15.86	4.99	9.19	13.17
600	9.0E+00	61.00	25.00	72.0	8.59	17.92	23.62	7.32	16.50	19.32	78.0	7.28	15.19	20.02	6.24	14.07	16.48
696	9.0E-07	33.00	25.00	84.0	6.22	10.04	17.11	5.37	9.13	14.17	92.0	5.10	8.24	14.04	4.44	7.55	11.71
768	9.0E-07	35.00	25.00	83.0	6.38	10.89	17.56	5.50	9.97	14.52	91.0	5.23	8.92	14.38	4.54	8.23	11.98
840	9.0E-07	30.00	25.00	84.0	6.22	11.01	17.11	5.37	10.11	14.17	91.0	5.23	9.25	14.38	4.54	8.56	11.98
936	9.0E-07	30.00	25.00	82.0	6.55	10.79	18.02	5.64	9.84	14.89	89.0	5.49	9.04	15.10	4.76	8.30	12.56
1008	9.0E-07	31.00	25.00	85.0	6.06	10.61	16.68	5.24	9.75	13.82	92.0	5.10	8.93	14.04	4.44	8.26	11.71
1104	9.0E-07	28.00	25.00	85.0	6.06	10.61	16.68	5.24	9.75	13.82	92.0	5.10	8.93	14.04	4.44	8.26	11.71
1200	9.0E-07	22.00	25.00	84.0	6.22	10.10	17.11	5.37	9.21	14.17	91.0	5.23	8.49	14.38	4.54	7.79	11.98
1272	9.0E-07	23.00	25.00	85.0	6.06	10.97	16.68	5.24	10.10	13.82	92.0	5.10	9.23	14.04	4.44	8.56	11.71
1344	9.0E-07	28.00	25.00	85.0	6.06	10.30	16.68	5.24	9.40	13.82	92.0	5.10	8.67	14.04	4.44	7.96	11.71
1416	9.0E-07	28.00	25.00	85.0	6.06	10.30	16.68	5.24	9.40	13.82	92.0	5.10	8.67	14.04	4.44	7.96	11.71
1536	9.0E-07	24.00	25.00	92.0	5.10	12.43	14.04	4.44	11.68	11.71	98.0	4.44	10.80	12.20	3.88	10.21	10.23
1608	2.0E-07	26.00	25.00	84.0	6.22	9.80	17.11	5.37	8.91	14.17	92.0	5.10	8.04	14.04	4.44	7.37	11.71
1680	2.0E-07	16.00	25.00	84.0	6.22	10.38	17.11	5.37	9.49	14.17	92.0	5.10	8.52	14.04	4.44	7.84	11.71
1776	2.0E-07	24.00	25.00	84.0	6.22	10.15	17.11	5.37	9.24	14.17	92.0	5.10	8.32	14.04	4.44	7.64	11.71
1848	2.0E-07	19.00	25.00	81.0	6.72	11.35	18.49	5.78	10.37	15.26	89.0	5.49	9.27	15.10	4.76	8.53	12.56
1944	2.0E-07	18.00	25.00	85.0	6.06	10.22	16.68	5.24	9.36	13.82	91.0	5.23	8.81	14.38	4.54	8.11	11.98
2016	2.0E-07	19.00	25.00	84.0	6.22	9.80	17.11	5.37	8.91	14.17	92.0	5.10	8.04	14.04	4.44	7.37	11.71
2112	2.0E-07	26.00	25.00	85.0	6.06	10.50	16.68	5.24	9.63	13.82	92.0	5.10	8.83	14.04	4.44	8.16	11.71

ENDURANCE TEST NO. 3
OIL = RBRC 36233
BEARING ROUGHNESS = DOUBL
INITIAL OIL FILM THICKNESS = THICK
LOAD(LB) = 200
SPEED(RPM) = 100
CENTRAL OUTER CONTACTS = CENTRAL INNER CONTACTS MULTIPLIED BY 1.08432
MINIMUM OUTER CONTACTS = MINIMUM INNER CONTACTS MULTIPLIED BY 1.10109

FILM THICKNESSES ARE IN MICROINCHES

TEST TIME (HR)	PRESSURE (TORR)	TORQUE (OZ-IN)	BASE SPEED (RPM)	AFT BEARING, INNER CONTACTS: TEMPA (F)	BHCAI	XCENI	TCENI	BHMAI	XMINI	TMINI	FORWARD BEARING, INNER CONTACTS: TEMPF (F)	BHCFI	XCENI	TCENI	BHMFI	XMINI	TMINI
2184	2.0E-07	26. 0	25.00	84.0	6.22	10.85	17.11	5.37	9.92	14.17	92.0	5.10	8.90	14.04	4.44	8.20	11.71
2280	2.0E-07	20.00	25.00	83.0	6.38	10.16	17.56	5.50	9.22	14.52	90.0	5.36	8.53	14.73	4.65	7.79	12.27
2352	2.0E-07	19.00	25.00	84.0	6.22	10.06	17.11	5.37	9.17	14.17	90.0	5.36	8.66	14.73	4.65	7.94	12.27
2424	1.5E-07	18.00	25.00	85.0	6.06	10.30	16.69	5.24	9.40	13.82	92.0	5.10	8.67	14.04	4.44	7.96	11.71
2544	1.5E-07	21.00	25.00	83.0	6.38	10.89	17.56	5.50	9.97	14.52	91.0	5.23	8.92	14.38	4.54	8.23	11.98
2616	1.5E-07	23.00	25.00	84.0	6.22	9.98	17.11	5.37	9.09	14.17	91.0	5.23	8.39	14.38	4.54	7.69	11.98
2688	1.5E-07	27.00	25.00	78.0	7.28	11.91	20.02	6.24	10.83	16.48	86.0	5.91	9.67	16.27	5.11	8.87	13.49
2784	1.5E-07	13.00	25.00	81.0	6.72	11.39	18.49	5.78	10.37	15.26	92.0	5.10	8.65	14.04	4.44	7.96	11.71
2856	1.5E-07	12.00	25.00	84.0	6.22	10.73	17.11	5.37	9.80	14.17	92.0	5.10	8.80	14.04	4.44	8.10	11.71
2952	1.5E-07	16.00	25.00	81.0	6.72	11.19	18.49	5.78	10.18	15.26	88.0	5.62	9.37	15.47	4.87	8.58	12.86
3072	1.5E-07	21.00	25.00	83.0	6.38	10.22	17.56	5.50	9.30	14.52	89.0	5.49	8.79	15.10	4.76	8.04	12.56
3192	1.5E-07	13.00	25.00	83.0	6.38	10.78	17.56	5.50	9.86	14.52	89.0	5.49	9.27	15.10	4.76	8.53	12.56
3288	1.5E-07	18.00	25.00	85.0	6.06	9.34	16.68	5.24	8.47	13.82	91.0	5.23	8.05	14.38	4.54	7.34	11.98
3408	1.5E-07	16.00	25.00	86.0	5.91	9.42	16.27	5.11	8.58	13.49	92.0	5.10	8.13	14.04	4.44	7.45	11.71
3528	1.5E-07	13.00	25.00	85.0	6.06	10.50	16.68	5.24	9.63	13.82	92.0	5.10	8.83	14.04	4.44	8.16	11.71
3624	1.5E-07	23.00	25.00	85.0	6.06	9.61	16.68	5.24	8.74	13.82	92.0	5.10	8.09	14.04	4.44	7.41	11.71
3696	1.5E-07	16.00	25.00	84.0	6.22	10.46	17.11	5.37	9.53	14.17	91.0	5.23	8.79	14.38	4.54	8.06	11.98
3742	1.5E-07	18.00	25.00	84.0	6.22	10.46	17.11	5.37	9.53	14.17	91.0	5.23	8.79	14.38	4.54	8.06	11.98
3864	1.5E-07	13.00	25.00	84.0	6.22	10.77	17.11	5.37	9.88	14.17	91.0	5.23	9.05	14.38	4.54	8.36	11.98
3960	1.5E-07	12.00	25.00	85.0	6.06	9.37	16.68	5.24	8.51	13.82	92.0	5.10	7.89	14.04	4.44	7.21	11.71
4032	1.2E-07	9.00	25.00	80.0	6.90	11.10	18.99	5.93	10.09	15.65	89.0	5.49	8.83	15.10	4.76	8.10	12.56
4128	1.2E-07	12.00	25.00	84.0	6.22	9.80	17.11	5.37	8.91	14.17	92.0	5.10	8.04	14.04	4.44	7.37	11.71
4224	1.2E-07	14.00	25.00	85.0	6.06	10.07	16.68	5.24	9.20	13.82	92.0	5.10	8.47	14.04	4.44	7.80	11.71
4296	1.2E-07	8.00	25.00	84.0	6.22	9.53	17.11	5.37	8.64	14.17	91.0	5.23	8.01	14.38	4.54	7.31	11.98

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